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THESIS

AN EXPERIMENTAL APPARATUS
TO STUDY NUCLEATE POOL BOILING OF
R-114 AND OIL MIXTURES

by

Mustafa Karasabun

December 1984

Thesis Advisor: P. J. Marto

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#20 ABSTRACT (Continued)

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An Experimental Apparatus to Study Nucleate Pool Boiling of R-114 and Oil Mixtures

by

Mustafa Karasabun
LTJG, Turkish Navy
B.S., Turkish Naval Academy, 1978

Submitted in partial fulfillment of the
requirements for the degree of

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In order to study the nucleate pool-boiling performance of R-114 and R-114-oil mixtures from enhanced evaporator tube surfaces, an experimental apparatus was designed, constructed and instrumented. The evaporator was made of a T-shaped Pyrex glass container. Boiling occurred from a smooth, hard-copper tube, 15.9 mm (5/8 in.) in outer diameter, 12.7 mm (1/2 in.) in inside diameter and 431.8 mm (17 in.) in length. The tube was heated using a cartridge heater, and it was instrumented with 8 thermocouples to measure the wall temperature. A Hewlett-Packard 3497A data acquisition/control unit and a 9826A computer were used to collect and process data. The condenser was cooled by an ethylene glycol-water mixture, which was maintained at about -17 °C by means of an R-12 refrigeration system. Nine data runs were completed to de-bug the experimental apparatus and to check for reproducibility. During all data runs, especially at higher heat fluxes (greater than 10 kW/m^2), large temperature variations were observed along and around the active boiling length of the test tube. The data were compared with data found in the literature and reasonable agreement was obtained.

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NOMENCLATURE

A	area
A_b	tube outside surface area of active boiling section
A_c	cross-sectional area of tube
A_s	tube outside surface area of non-boilng section
$a_0 \ldots a_7$	conversion coefficients for thermocouple
C_p	specific heat
D	diameter
D_i	tube inside diameter
D_o	tube outside diameter
D_1	diameter at the position of the thermocouple
D_2	outer diameter of the boiling tube
DCP	discrepancy
E	thermocouple reading
g	gravitational acceleration
h	heat-transfer coefficient
h_{fg}	latent heat of vaporization
I	current
I_s	current reading by AC Current Sensor
k	thermal conductivity of liquid
k_c	thermal conductivity of copper
L	active boiling tube length
L_u	non-boiling length of the test tube
Nu	Nusselt number
P	pressure
P_c	critical pressure
p	tube outside wall perimeter
Pr	Prandtl number
Q	heat-transfer rate from boiling surface
Q_F	heat-transfer rate through one unboiling end
Q_H	heat-transfer rate from cartridge heater
Ra	Rayleigh number

T	temperature
T_{avg}	average wall temperature at the thermocouple location
T_b	calibration bath temperature
T_n	temperature of the thermocouple location
T_{sat}	saturation temperature of the boiling liquid
T_{wo}	outer wall temperature of the boiling test tube
V	voltage across the cartridge heater
V_s	voltage reading by AC-DC true RMS converter
α	thermal diffusivity
β	volumetric thermal expansion coefficient
δ	uncertainty in measurement and calibration
μ	viscosity
ν	kinematic viscosity
Φ	volume fraction of pure refrigerant
ρ_L	saturated liquid density
ρ_V	saturated vapor density

I. INTRODUCTION

A. BACKGROUND

In order to reduce the size and weight of air-conditioning systems and water chillers, the United States Navy recently has considered two different approaches: 1) to change the type of the refrigerant from R-11 to R-114, and 2) to use enhanced evaporator tube surfaces, such as Gewa-T, Thermoexcel-E, and High-Flux. Schematic drawings and photomicrographs of these enhanced surfaces at various magnifications are given in Figures 1.1 , 1.2 and 1.3. A combination of R-114 and enhanced surfaces may lead to significant improvements in system efficiency as well as to smaller and lighter components in future Navy applications.

B. ADVANTAGES OF USING R-114

The latent heat per unit volume of R-114 is 50 BTU/ft^3 at 70 °F. As a result, the heat carried by each cubic foot of R-114 is nearly twice as much as that carried by R-11 [Ref. 1] at the same temperature. In other words, the volume of R-114 vapor circulated per ton of refrigeration is about half that for R-11. Inherently, the size of the system components can be reduced significantly using R-114 rather than R-11. R-12 and R-22 in fact can transfer the heat more efficiently than R-114, but they require higher pressure ranges. However, operating pressures for R-114 are moderate, similar to R-11, such that light-weight components may be desirable and less-experienced personnel may be involved in the installation and maintenance. While the refrigerants in general are nonflammable, R-114 and R-12 are rated to have the lowest toxicity among all refrigerants [Ref. 2]. Based on the above discussion, R-114 is probably the most suitable refrigerant for use on board submarines and ships.

Further, R-114 is considerably more stable toward heat than R-12. All standard ferrous and nonferrous metals, except zinc and magnesium, and aluminum alloys containing appreciable amounts of zinc and magnesium, may be used with R-114 [Ref. 3]. The physical properties of R-114 and its comparison with the other common refrigerants are summarized in Appendix A, pressure-enthalpy diagram of R-114 is provided in Appendix B and the range of application of refrigerants are provided in Appendix C [Ref. 4].

C. ENHANCED BOILING TUBE SURFACES

The evalution of enhanced surface geometries that promote high-performance nucleate boiling has been surveyed by Webb [Ref. 5]. Two basic types of enhanced boiling surfaces are employed on the outer surface of tubes: porous coatings and reentrant grooves. The High-Flux surface is a sintered, porous metallic matrix bonded to the base surface. The Gewa-T and Thermoexcel-E surfaces have reentrant-type nucleation sites formed by cold working an integral finned tube wall. See Figures 1.1, 1.2 and 1.3 for details.

The key to the high performance of these surfaces is attributed to three factors [Ref. 5] : "(1) a pore or reentrant cavity within a critical size range, (2) interconnected cavities, and (3) nucleation sites of a reentrant shape. When the cavities are interconnected, one active cavity can activate adjacent cavities. It appears that the dominant fraction of the vaporization occurs at a very thin liquid film within the subsurface structure. The reentrant cavity shape provides a stable vapor trap, which will remain active at very low liquid superheat values."

The pool boiling heat-transfer performance of R-113 from the above-mentioned commercially-available enhanced surfaces has been investigated by Marto and Lepere [Ref. 6]. They reported that for R-113, the enhanced surfaces showed a two to tenfold increase in the heat-transfer coefficient when compared to a plain tube. The High-Flux surface was most effective over a broad range of heat fluxes, whereas the Gewa-T surface was only effective at high heat fluxes (near 100 kW/m²). The Thermoexcel-E surface showed similar gains in the heat-transfer coefficient to that of the High-Flux surface below 10 kW/m². They also reported that the degree of superheat required to activate the enhanced surfaces was less than the plain tube, and was sensitive to initial surface conditioning and fluid properties.

D. NUCLEATE POOL BOILING PERFORMANCE OF PURE R-114

The heat-transfer performance of pure R-114 in the evaporator section of an air-conditioning unit can be estimated using one of the suggested commonly-used correlations for nucleate pool boiling, which are listed by Chongrungreong and Sauer [Ref. 7]. However, because of the lack of complete understanding of the nucleate boiling mechanism, these correlations should be compared with experimental data if it is possible. So far, the literature reveals a limited amount of nucleate boiling experimental investigations using R-114. Happel [Ref. 8], using a horizontal, pure-nickel tube with a surface roughness of 0.43 micrometer, investigated the boiling characteristics of pure R-114 at various pressures. The effect of pressure on nucleate boiling heat transfer in R-114 was also investigated experimentally by Nishikawa and his co-workers [Ref. 9]. They operated over a reduced pressure (P/P_c) range from 0.03 to 0.98, and demonstrated that the heat transfer coefficient of R-114 increases remarkably at high pressure near the critical point. Their results showed that the effect of surface roughness changes with pressure and the effect gradually vanishes as pressure approaches the critical point. Using a horizontal smooth copper tube, Henrici [Ref. 10] investigated boiling heat-transfer performance of R-114 over a heat-flux range from 0.1 to 100 kW/m^2 and over a pressure range from 37 to 252 kN/m^2.

Stephan and Mitrovic [Ref. 11] investigated the performance of pure R-114 from a Gewa-T enhanced surface. Boiling occurred outside a horizontal bundle of these tubes. They reported that the position of the tube within the bundle does not effect the heat-transfer coefficient considerably. The reason for this effect was explained in the following manner: "The flow pattern around a tube seems to be of very weak influence and the heat-transfer process is mainly governed by the bubble formation and the two-phase flow inside the T-shaped channel. The top part of the fins, which were rolled to form the T, practically did not directly contribute to the heat-transfer process because, even at maximum heat fluxes, vapour bubbles were not produced there."

Marto and Hernandez [Ref. 12] also investigated nucleate pool-boiling characteristics of a Gewa-T surface in R-113. Their experimental results showed a disagreement with Stephan and Mitrovic's above explanation. They showed that "the liquid-vapor motion may not always be confined to flow circumferentially within the channels formed by the T-shaped fins as previously postulated. Surface imperfections on the tips of the Gewa-T fins may cause bubble nucleation from these sites. The presence of these active sites may further enhance the performance of the Gewa-T surface."

The factors which appear to affect boiling heat-transfer performance of a single tube are the physical properties of refrigerant, boiling pressure, tube diameter, surface condition of the tube, and maybe the "submergence effect" (resulting from the hydrostatic liquid head) on the evaporation process. Experimental observations have shown that changes in magnitude of these properties and conditions significantly affect pool-boiling heat-transfer performance. [Ref. 7].

E. NUCLEATE BOILING PERFORMANCE OF REFRIGERANT-OIL MIXTURES

Most of the refrigeration systems work with oil-lubricated, hermetically sealed compressors, which are cooled by the working fluid. Therefore, the fluids being circulated within these refrigeration systems generally are not pure refrigerants, but are refrigerant-oil mixtures. The saturation temperature of a refrigerant-oil mixture is higher than that for the pure refrigerant at the same pressure. Also, both viscosity and surface tension increase when oil is added to refrigerants. Other thermodynamic properties of refrigerants, such as density and specific heat, are also significantly changed by the addition of oil. Thus, adequate knowledge on the influence of oil on the boiling of refrigerants is very important for the successful design of refrigeration systems. The literature reveals only very little information on the boiling performance of R-114-oil mixtures [Ref. 10]. In particular, heat-transfer performance of the enhanced evaporator surfaces in refrigerant-oil mixtures is a poorly-understood subject area. Since these advanced boiling surfaces contain a high density of pores, the presence of oil can generally result in a greater performance degradation than on more-conventional, smooth surfaces.

Various studies which are surveyed and listed by Jensen and Jackman [Ref. 13] have shown that, in general, nucleate boiling heat-transfer coefficients decrease with increasing oil concentration in R-11, R-12, R-22, R-113 and R-114. For oil concentrations less than 3%, however, Dougherty and Sauer [Ref. 14] demonstrated that the heat-tansfer coefficient may be increased slightly for R-11-oil and R-113-oil mixtures. They attributed this trend to an increase of foaming of the mixtures. This result also has been confirmed by Sauer, Gibson and Chongrungreong [Ref. 15] using R-12-oil mixtures, and by Henrici [Ref. 10] using R-114-oil mixtures. However, in a recent experimental study using R-113-oil mixtures, Jensen and Jackman [Ref. 13] did not observe such an enhancement with low oil concentrations. The model of the governing processes involved in boiling of refrigerant-oil mixtures has been given by Stephan [Ref. 16] as follows: "Obviously the reduction in heat transfer in a mixture is due to the fact that the liquid near the vapour-liquid interface of growing bubbles becomes denuded in the more volatile components. Thus, a concentration gradient exists between the interface and the bulk of the liquid, and the more volatile components have to diffuse through the liquid in order to reach the interface. A mass-transfer resistance thus is introduced, which is not present when boiling single components."

The effect of the oil viscosity on heat transfer in boiling of refrigerant-oil mixtures has been considered by Sauer, Gibson, and Chongrungreong [Ref. 15]. Oil viscosity is believed to significantly contribute to the changes in boiling characteristics of refrigerant-oil mixtures. The heat-transfer coefficient is greatest for the highest viscosity of oil for the same temperature difference.

The effect of oil on boiling of R-12 from a Gewa-T tube was investigated by Stephan and Mitrovic [Ref. 17]. They reported that the presence of oil in R-12 reduced the heat-transfer coefficient almost over the entire range of heat fluxes and mass fractions. They observed an increase in heat-transfer coefficient with less than 5% mass fraction of oil only at high heat fluxes (such as higher than 20 kW/m^2).

The effects of oil on the heat-transfer performance of refrigerants in a tube bundle have been summarized by Arai, et al. [Ref. 18] in the following manner: "The presence of oil in a shell-and-tube evaporator is believed to have two opposing effects on the heat-transfer coefficient of the evaporator tube bundles. Firstly, many people suspect that the oil left out from the evaporating refrigerant tends to choke the surface structure, thereby lowering the heat-transfer coefficient. Secondly, from the experience of conventional heat exchangers, it has long been recognized that the presence of oil enhances foaming on the upper part of the tube bundles. This foaming increases the vigor of refrigerant movement around the tubes, so that heat transfer is enhanced."

F. THESIS OBJECTIVES

Based on the foregoing discussion, it is clear that little information exists in the literature to enable the successful design of R-114 evaporators, especially when using enhanced boiling surfaces, in the presence of lubricating oils. The objectives of this research effort were therefore to:

1. Design, construct and instrument an experimental apparatus in order to investigate the heat-transfer performance of R-114 and R-114-oil mixtures from enhanced evaporator tube surfaces,
2. Successfully operate the system to produce repeatable data, and
3. Obtain smooth-tube baseline data for R-114 to be used as a standard for the future data on advanced surfaces.

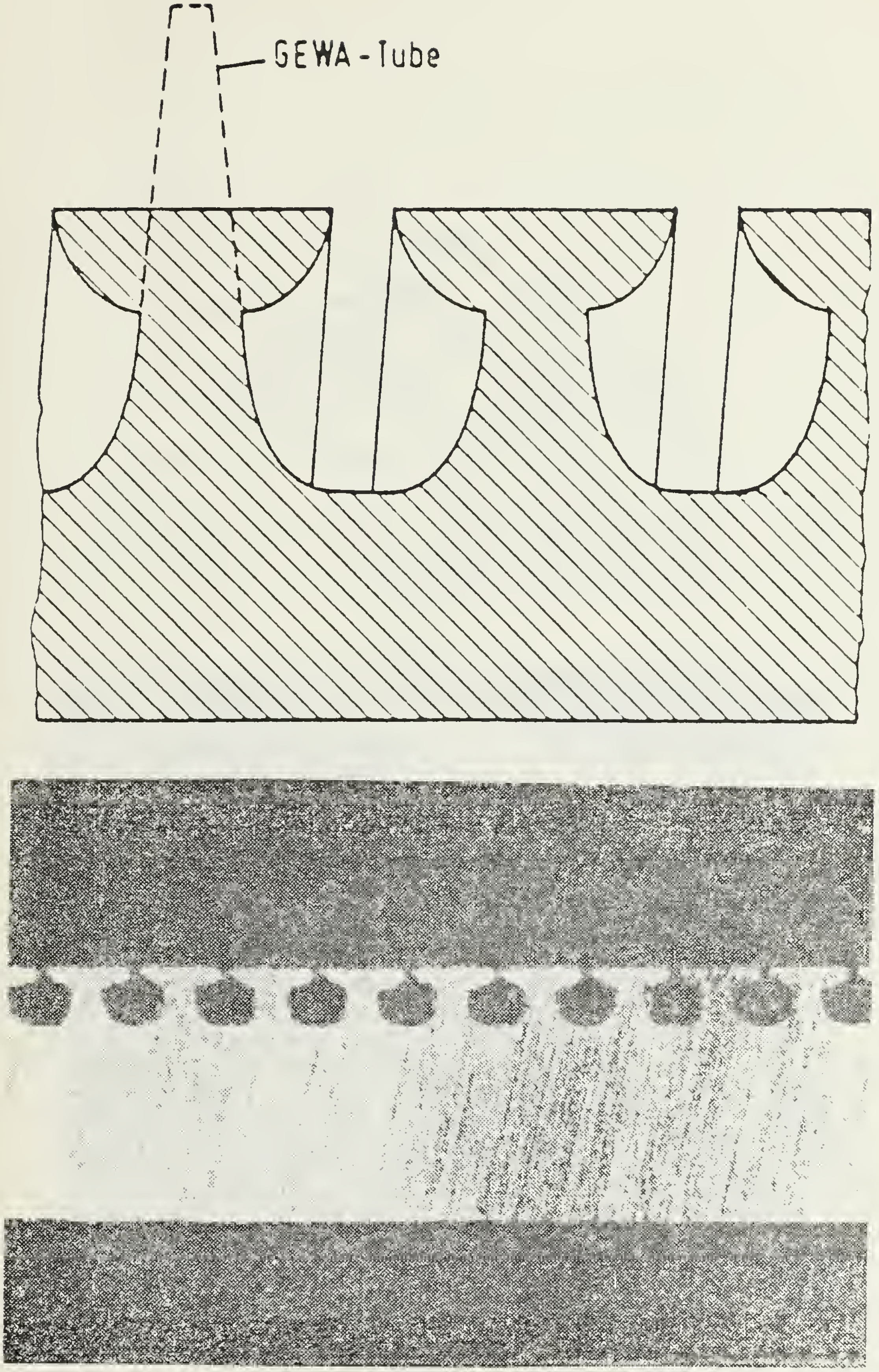


Figure 1.1 Schematic and Optical Micrograph of a Cross Section of the Gewa-T Surface (20X).

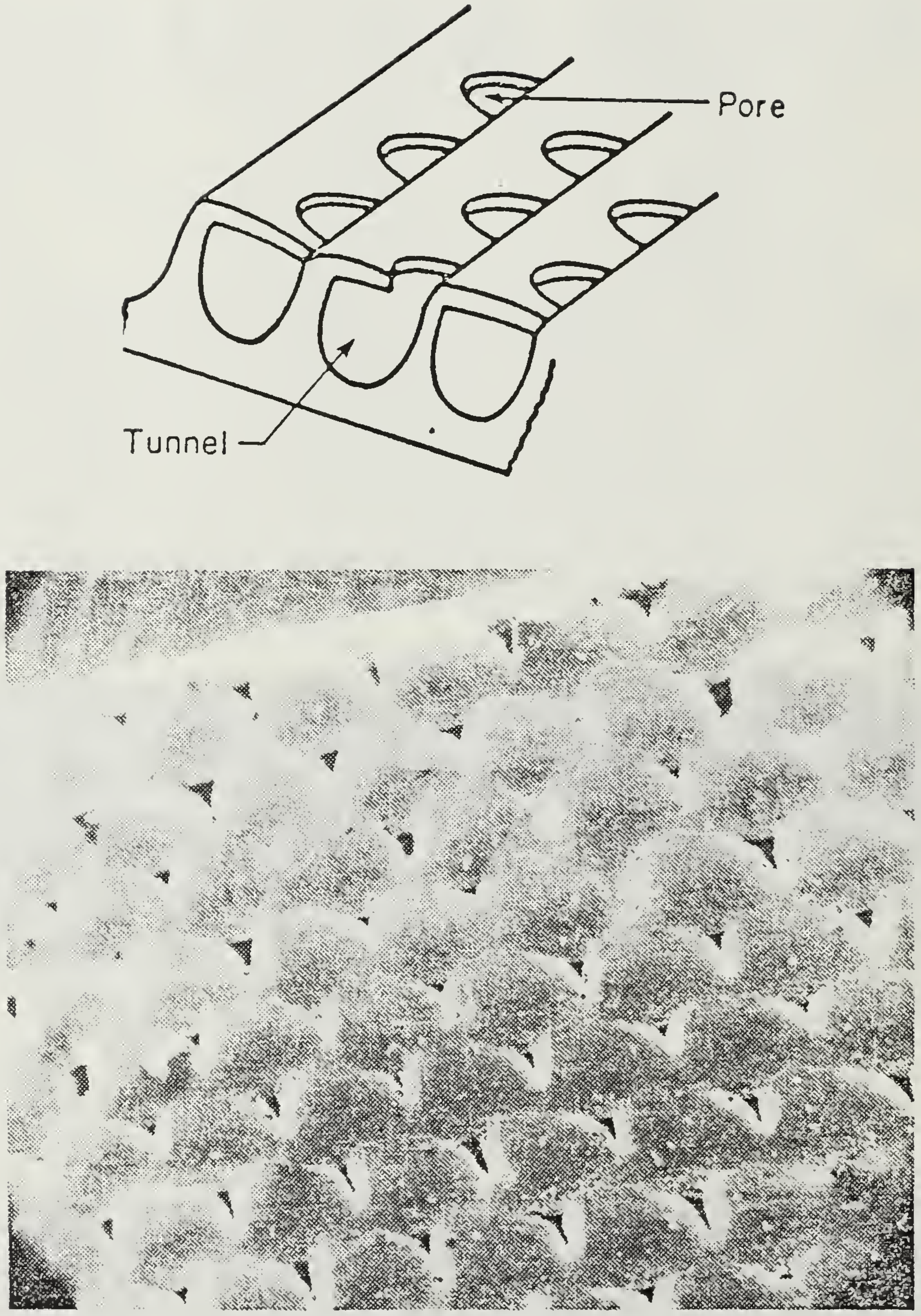


Figure 1.2 Schematic and Optical Micrograph of the Thermoexcel-E Surface (40X).

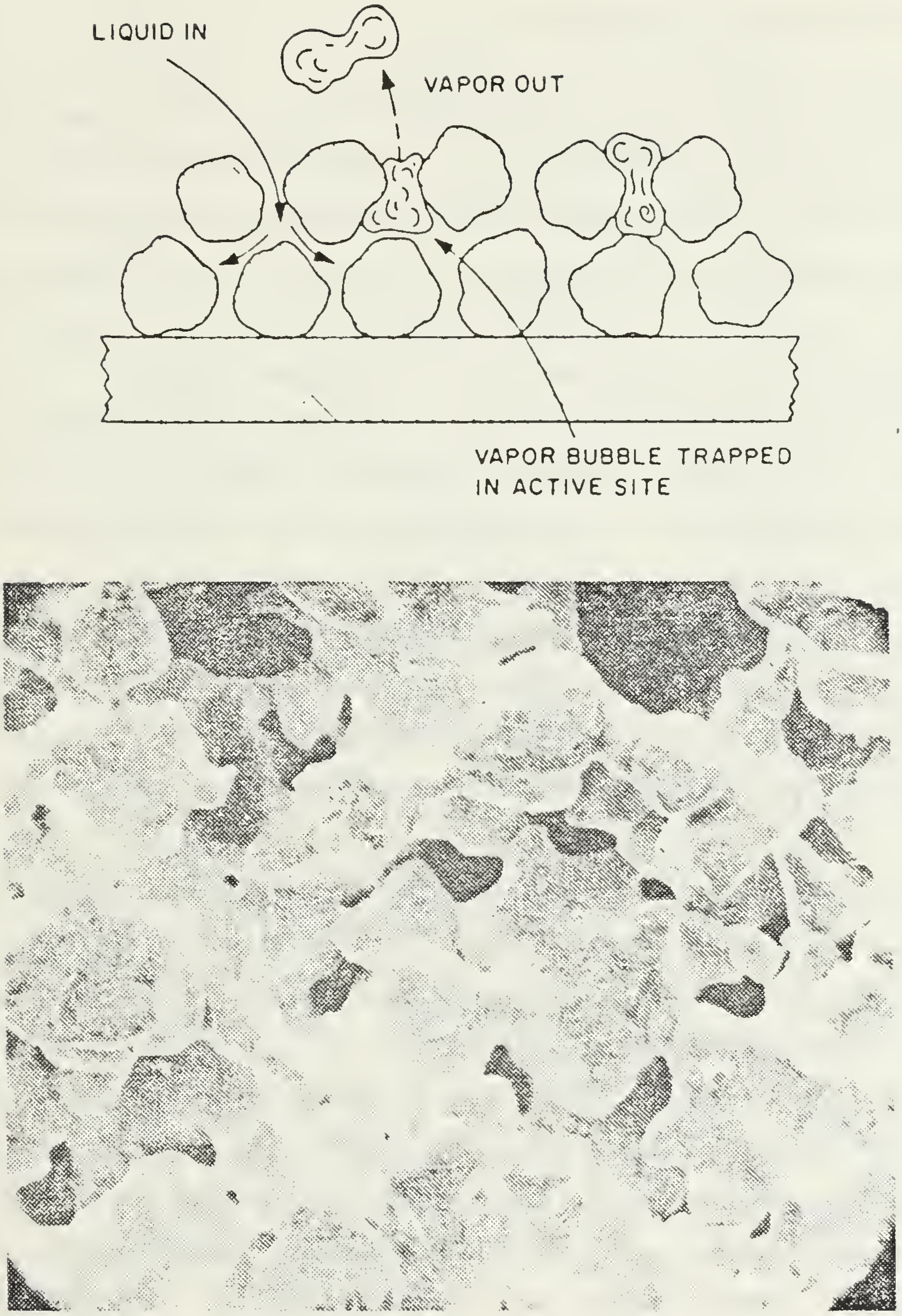


Figure 1.3 Schematic and Scanning Electron Micrograph of the High Flux Surface (500X).

II. DESCRIPTION OF EXPERIMENTAL APPARATUS

A. OVERVIEW OF THE SYSTEM

An overall schematic representation of the experimental apparatus is shown in Figure 2.1, and a photograph is shown in Figure 2.2. The apparatus consists of a Pyrex-glass evaporator with the boiling test tube 1, a Pyrex-glass condenser 2, an R-114 liquid reservoir 3, an oil reservoir 4, a graduated oil cylinder 5, a vacuum pump 6, a cooling section consisting of a 1/2-Ton R-12 refrigeration unit, a tank filled with water-ethylene glycol mixture 7, and a pump 8. The evaporator 1, condenser 2, oil-reservoir 4, graduated-oil cylinder 5 and R-114 reservoir 3, are placed in a Plexiglas chamber 9, where the temperature can be maintained at lower values (up to 10 °C) relative to ambient temperature. This chamber also served as a safety barrier in case of an inadvertent overpressurization of the glass apparatus.

The vapor rising from the evaporator is condensed in the condenser 2 and the liquid is then fed back into the evaporator through valve V5. In this manner, the specific enthalpy of condensation ("latent heat") of R-114 vapor is removed in the condenser. The evaporator of the R-12 refrigeration system is made of a copper coil, which is immersed in the water-ethylene glycol tank in order to maintain the temperature of the mixture at about -17 °C. An 8-GPM, turbine-type pump 8 transports the water-ethylene glycol mixture through the control valve VC to the condenser.

The oil can be added into the evaporator 1 using the graduated cylinder 5 by opening valves V1 and V4. The graduated cylinder is supplied with oil by reservoir 4. The valves V5, V6 and V7 are provided for emptying the evaporator prior to dismantling it, for example, for changing the evaporator tube. Also, fresh R-114 can be added to the reservoir while valve V5 is closed and V6, V7 are open and by providing R-114 vapor through V8 and V10.

The test tube was placed horizontally in the evaporator after a 240 VAC electrical cartridge heater of 203.2 mm (8 in.) in length had been inserted in its middle section. The whole system was connected to the vacuum pump through V8 and V10 in order to remove noncondensible gases. The relief valve was set to a gage pressure limit of 138 kN/m^2 (20 psi), which is 50% less than the manufacturer-recommended working pressure limit of the Pyrex glass tees. All connections were assembled with Swagelok fittings sealed with Teflon ferrules to ensure leak tightness. Also, all valves used in the system, except control valve VC and by-pass valve V9, were made of stainless steel. Copper tubes of 9.5-mm (3/8 in.) diameter were used for all piping, except in the oil section, where 6.35-mm (1/4 in.) tubes were used.

B. BOILING TEST SECTION

1. Evaporator

The evaporator is a T-shaped container made of Corning Pyrex glass. Figure 2.3 shows a schematic of the evaporator with dimensions. The selection of Pyrex glass offers several advantages: it is corrosion-resistant and transparent and has a smooth interior surface (this minimized nucleate boiling at the inner surface of the container) compared to metals, and is more stable with temperature and pressure variations compared to ordinary glass. The operating pressure limit of the evaporator is guaranteed by the manufacturer up to a gage pressure of 200

kN/m^2 (30 psi). Each end of the Pyrex-glass evaporator was fitted with a cast-iron flange and a gasket. A detailed sketch of the cast-iron flange is shown in Figure 2.3. Two aluminum flanges, 210 mm in diameter and 12.7 mm in thickness, were bolted to the cast-iron flanges. Thus, all fittings were connected through the aluminum flanges to the Pyrex-glass evaporator.

2. Test Section

A schematic drawing of the test section is shown in Figure 2.4. The boiling test tube was held in place by two Teflon bushings, which were attached to the aluminum flanges at both ends of the T-shaped evaporator. Four studs were used in order to attach the Teflon bushing on the aluminum flange. The Teflon bushing and both ends of the test tube were sealed by means of two rubber O-rings in order to achieve leak tightness. An additional O-ring was placed between the aluminum flange and the Teflon plug at each end.

3. Boiling Tube

A smooth, hard-copper tube, 15.9 mm (5/8 in.) in outer diameter, 12.7 mm (1/2 in.) in inside diameter and 431.8 mm (17 in.) in length, was used to provide baseline data for the nucleate pool boiling heat-transfer coefficient of pure R-114. The heater was a 1000-Watt 240-Volt stainless-steel cartridge, 6.35 mm (1/4 in.) in outer diameter and 203.2 mm (8 in.) in length. The cartridge heater was inserted into a copper sleeve, which was 6.35 mm (1/4 in.) in inside diameter, 12.7 mm (1/2 in.) in outside diameter and 203.2 mm (8 in.) in length. In order to provide a uniform heat flux, the cartridge heater and the copper sleeve were soldered together. Pulido [Ref. 19] showed that (using a similar type of heater-element construction) the heater element provides a uniform heat flux along the circumferential direction. The heater element used during this thesis was assumed to have similar characteristics. The cartridge heater and the copper sleeve were then inserted as a unit into the middle portion of the test tube using a tight mechanical fit. (Note: A shrink-fit process is essential to minimize the thermal contact resistance. However, owing to the time constraint in this thesis, the shrink-fit process was delayed and a slide fit with a clearance of less than 0.01 mm was used. Thus, the resulting tube was used mainly for the purpose of obtaining a successfully-operating apparatus). The active boiling length of the test tube was 203.2 mm (8 in.) in the middle portion of the boiling tube. In order to compute the actual average heat flux in the heated portion of the tube, a suitable correction was applied for the natural convection created at both ends. See Heat Flux Calculation section in Chapter 4 for details.

To measure the wall surface temperature of the boiling tube, 8 thermocouples were inserted into 8 grooves which were machined on the outside of the copper sleeve. As shown in Figure 2.5, these thermocouples were located at different axial and circumferential locations. Four thermocouples (1,2,5,6) were placed with 90-degree separation around the copper sleeve to measure the temperature distribution at the mid cross section of the active length of boiling tube. The thermocouple's wire was glued down at several places along the thermocouple's channel using Epoxy. The longitudinal temperature distribution on the active boiling section was also measured by the symmetrically-located 4 thermocouples (3,4,7,8). The exact locations of the thermocouples and dimensions of the thermocouple grooves are given in Figure 2.5. All thermocouple grooves were axially machined from the location of the thermocouple hot junctions to the nearest end of the copper sleeve.

C. CONDENSER SECTION

The condenser was also a T-shaped container made of Corning Pyrex glass. It was identical to the evaporator. The position of the condenser can be seen in Figure 2.1. R-114 was condensed on a vertical copper coil, which was inserted in the Pyrex-glass condenser. The copper coil was fabricated into a 76.2 mm (3 in.) diameter coil using 9.5 mm (3/8 in.) copper tube. The active condensation length was estimated to be 4.5 m (15 ft).

The top portion of the condenser was connected to a portable, mechanical vacuum pump to remove noncondensible gases from the apparatus. The bottom of the condenser was also connected to the evaporator via valve V5 in order to return the condensed R-114 liquid to the evaporator. The cooling liquid, i.e., water-ethylene glycol mixture, entered the top portion of the condenser, through the copper coil and left the condenser from the bottom to return to the water-ethylene glycol tank. The condenser was placed vertically and connected to the vapor outlet of the evaporator using L-shaped aliminum tube, 50.8 mm (2 in.) in diameter. The maximum vapor velocity of R-114 vapor through the aluminum tube was found to be about 0.5 m/sec. A Bourdon gage with a range of absolute vacuum to a gage pressure of 1030 kN/m^2 (150 psi) and a relief valve which was set to 138 kN/m^2 (20 psi) were placed on the L-shaped aluminum tube.

D. OIL ADDING SECTION

To study the boiling performance of R-114-oil mixtures, a cylindrical aluminum reservoir, 152.4 mm (6.in) in diameter and 152.4 cm (6 in.) in height, and a glass oil cylinder were installed above the evaporator. The relative positions of the oil reservoir and the oil cylinder are shown in Figure 2.1. The oil cylinder was 355 mm in length and had a diameter of 25.4 mm. This cylinder was specially ordered to achieve a resolution of 0.5 ml. The oil cylinder was connected to the oil reservoir through valves V3 and V2. The addition of oil into the evaporator can be achieved through V1 by gravity after balancing the pressure of the oil cylinder with that in the evaporator by opening valve V4.

E. COOLING SECTION

1. Water-Ethylene Glycol Mixture Tank

In order to store the water-ethylene glycol mixture, a special tank was manufactured. The total volume of the tank was 0.154 m^3 (0.48 m x 0.48 m x 0.66 m) and it was made of 12.7-mm-thick Plexiglas sheet. All sides of the tank were glued together with methylene-chloride solution. The joints were held together with small screws for extra strength. The low thermal conductivity of Plexiglas was especially suited to minimize heat transfer (from room to water-ethylene glycol mixture) through the tank walls. The tank was placed on the floor and all sides were insulated with 22 mm (7/8 in.) thick rubber insulation sheets. The cooling mixture contained 49 liters (13 gal.) of ethylene glycol and 94 liters of (25 gal.) distilled water. The freezing point of this mixture was about - 25 °C.

2. R-12 Refrigeration Plant

A 1/2-Ton R-12 refrigeration plant was installed to cool the water-ethylene glycol mixture. Figure 2.6 shows a schematic of the R-12 refrigeration plant. It consists of a compact-type air-cooled condenser, a compressor, a receiver, a filter-drier unit, a pressure regulator, a pressure-control switch and a thermostatic expansion valve. The evaporator of the R-12 refrigeration plant was constructed using a 9.5 mm (3/8 in.) copper tube, which was immersed in the water-ethylene glycol tank. The temperature of the water-ethylene glycol mixture is controlled by both a thermostatic expansion valve and a pressure control switch. The R-12 refrigeration plant was adjusted to keep the temperature of the cooling liquid at about -17 °C.

3. Pump and Control Valve

An 8 GPM, 115 VAC Burks turbine-type, positive-displacement pump was installed on the floor and the suction side of the pump, 25.4 mm (1 in.) in diameter, was directly coupled to the water-ethylene glycol tank. Cooling liquid was pumped from the tank to the condenser through the control valve VC. Also, a by-pass valve V9 was placed before the control valve VC on the discharge line. The use of the by-pass line served two important purposes: 1, it avoided overloading of pump 8 in the event value VC is completely closed, and 2, it provided proper mixing for the "warm" stream returning from the condenser 2. See Figure 2.1 for the positions of control valve VC and by-pass valve V9. The by-pass valve was adjusted and set permanently so that a sufficient range of flow rates can be achieved through valve VC to satisfy proper cooling of the condenser.

F. R-114 RESERVOIR

An aluminum cylindrical reservoir, 228 mm (9 in.) in diameter and 254 mm (10 in.) in height, was placed vertically between the evaporator and condenser in order to store R-114 as a liquid. The liquid level of the R-114 can be observed by means of a sight hose attached on the reservoir with proper fittings. R-114 reservoir was connected to the vapor line through valve V7 and to the liquid line through valve V6. See Figure 2.1 for arrangement of the reservoir.

G. CHAMBER

An aluminum frame (1.07 m x 0.51 m x 0.61 m) was constructed to locate all the parts of the apparatus, except the cooling section. All four vertical sides of the frame were covered with 12.7 mm (1/2 in.) thick Plexiglas sheets and both left and right sides were provided with hinges to enable easy access to the components of the apparatus. Aluminum and plywood plates were used to cover the bottom and top sides of the frame, respectively. The valve bodies of V1 through V8 were placed inside of the front Plexiglas sheet with the valve stems penetrating the sheet. Thus, the valve handles were accessible from outside of the Plexiglas box. The whole frame was placed above the water-ethylene glycol tank with aluminum support so that the system was very compact.

One of the main advantages of this chamber is that the temperature surrounding the evaporator can be reduced relative to the ambient temperature. Also, in case of emergency, the thick Plexiglas chamber would provide a safety barrier to personnel and equipment.

H. INSTRUMENTATION

1. Power Measurement

A 240 Volt AC source was used as the power supply, and it was adjusted by a variac in the range of 0-260 Volt and 0-8 Ampere according to the desired heat flux at the surface of the boiling tube. Power input to the boiling tube was measured with an AC current sensor and an AC-DC true R.M.S converter. The AC current sensor was connected to the input line of the heater in series and the AC-DC true R.M.S converter was connected in parallel. Figure 2.7 shows a schematic representation of the power-measurement devices.

Both the AC current sensor and the AC-DC true R.M.S converter were connected to the data acquisition/reduction system.

2. Temperature Measurement

Various temperatures were monitored throughout the system to include:

1. Boiling tube wall (8 thermocouples)
2. Liquid temperature (one thermocouple)
3. Vapor temperature (two thermocouples), and
4. Water-ethylene glycol mixture temperature (one thermocouple)

The locations of the wall thermocouples in the sleeve of the boiling tube are shown in Figure 2.5. The liquid and vapor thermocouples were inserted into the two specially-manufactured thermocouple wells. Figure 2.8 shows a schematic of these thermocouple wells. While the stainless-steel portion minimizes (owing to low thermal conductivity) errors resulting from the axial conduction of heat from the surrounding, the copper tip helps minimize the temperature drop from the area being measured to the thermocouple location (owing to the high thermal conductivity of copper).

All the temperature measurements were accomplished by 0.245 mm (30 gage) copper-constantan thermocouples. Each thermocouple measurement was read directly by a Hewlett-Packard 3497A data acquisition system, which was controlled by a Hewlett-Packard 9826 computer. Each thermocouple was scanned for 0.8 seconds and twenty readings were averaged to obtain a more accurate measurement.

A total of five thermocouples were calibrated. Two thermocouples were made from the beginning of a spool of copper-constantan wire; one was made from the mid portion and two were from the end. It was assumed that the properties of the copper and constantan do not change along a given section of wire for any given spool. All thermocouples were calibrated by the method described in Appendix D.

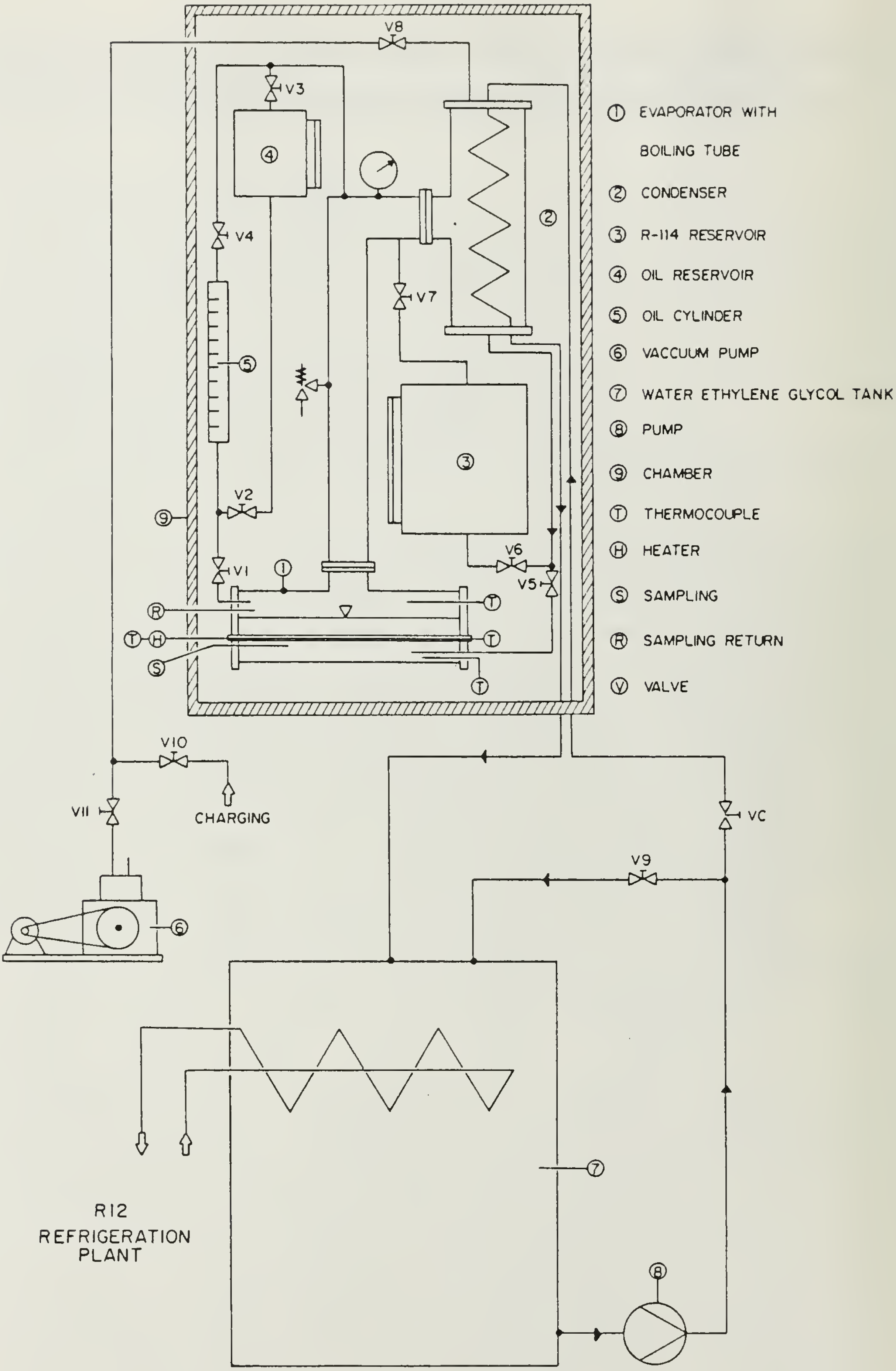


Figure 2.1 Schematic of the Boiling Test Apparatus.

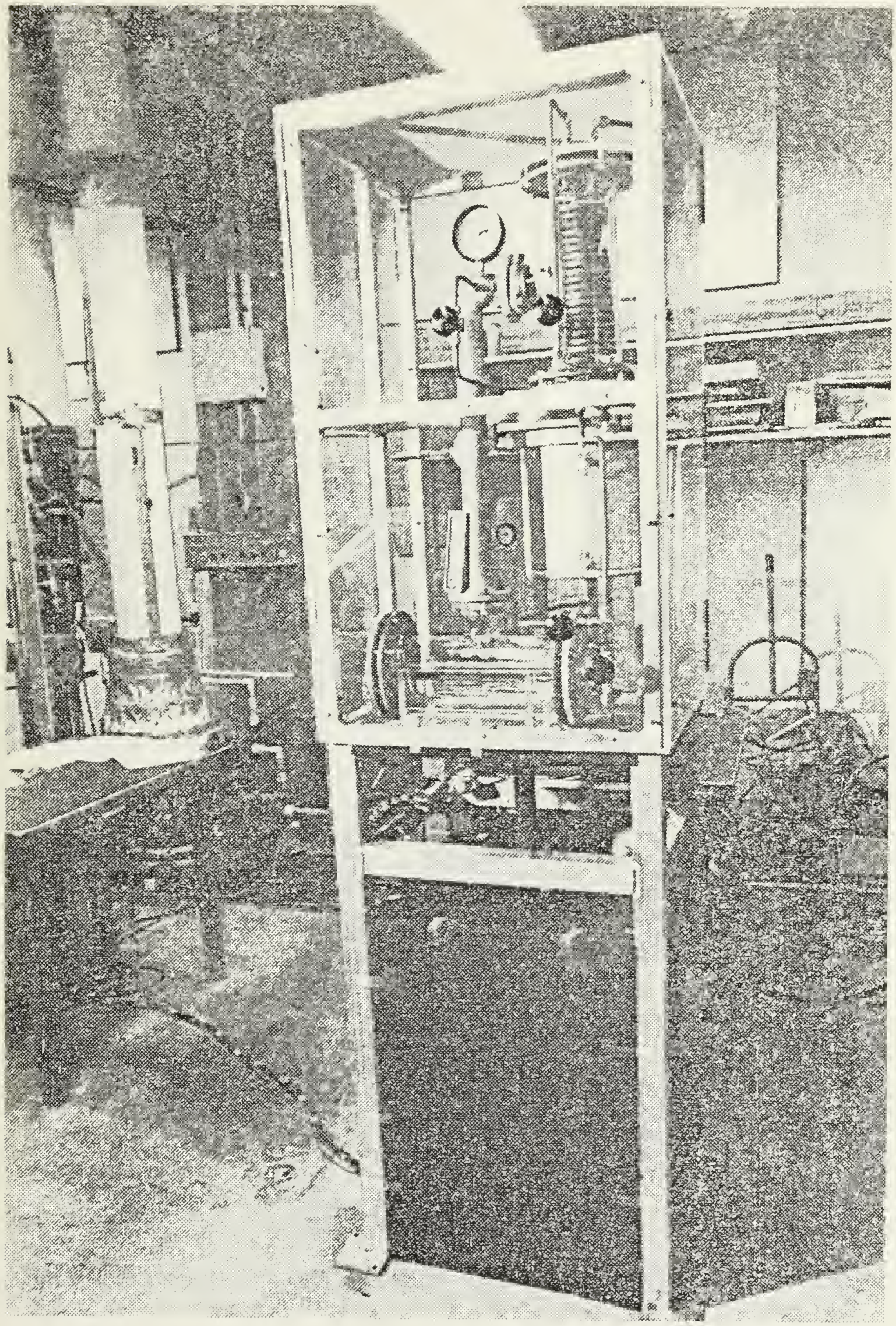
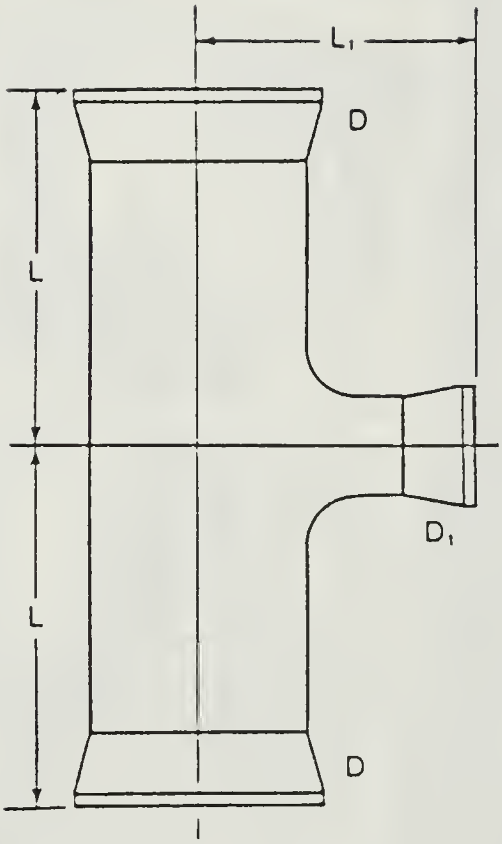
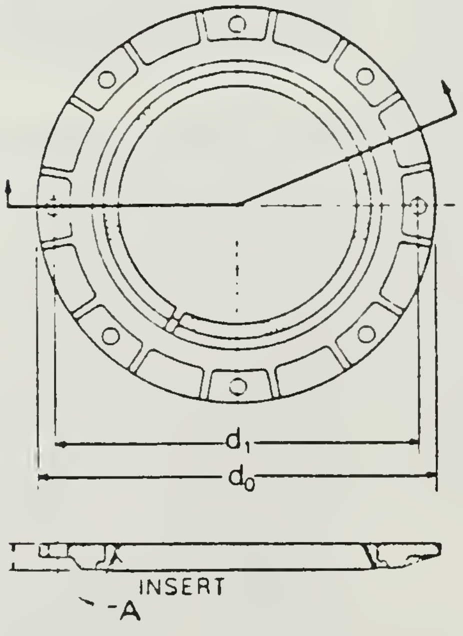


Figure 2.2 Photograph of Overall System.



a) Corning Pyrex Glass Evaporator (D x D_1 = 402x51 mm, L = 178 mm, L_1 = 127 mm)



b) Cast Iron Flange and Gasket (d_1 = 190mm, d_0 = 210mm, L_1 = 14mm, A = 21°)

Figure 2.3 Schematic of the Pyrex Glass Evaporator.

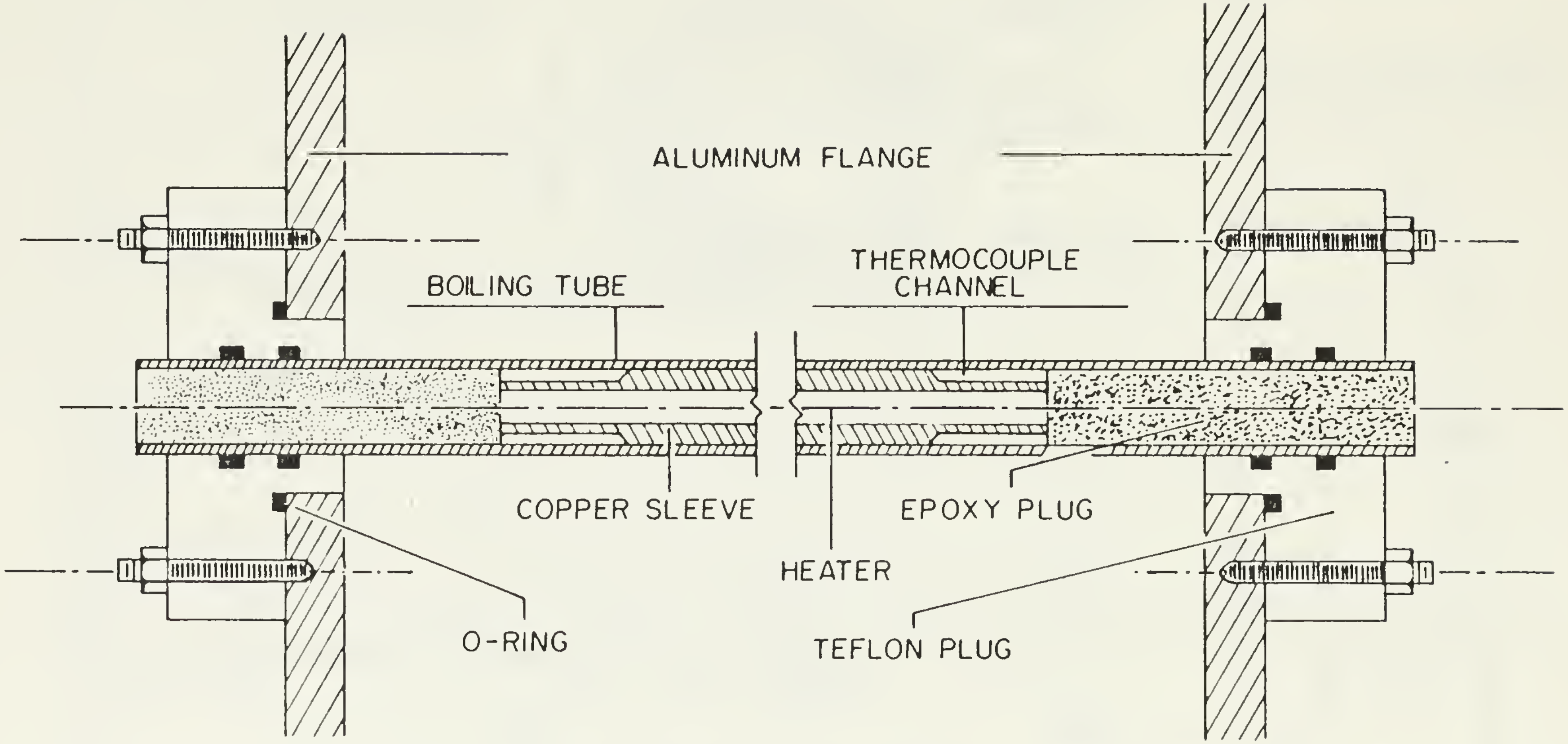


Figure 2.4 Schematic of the Boiling Test Tube.

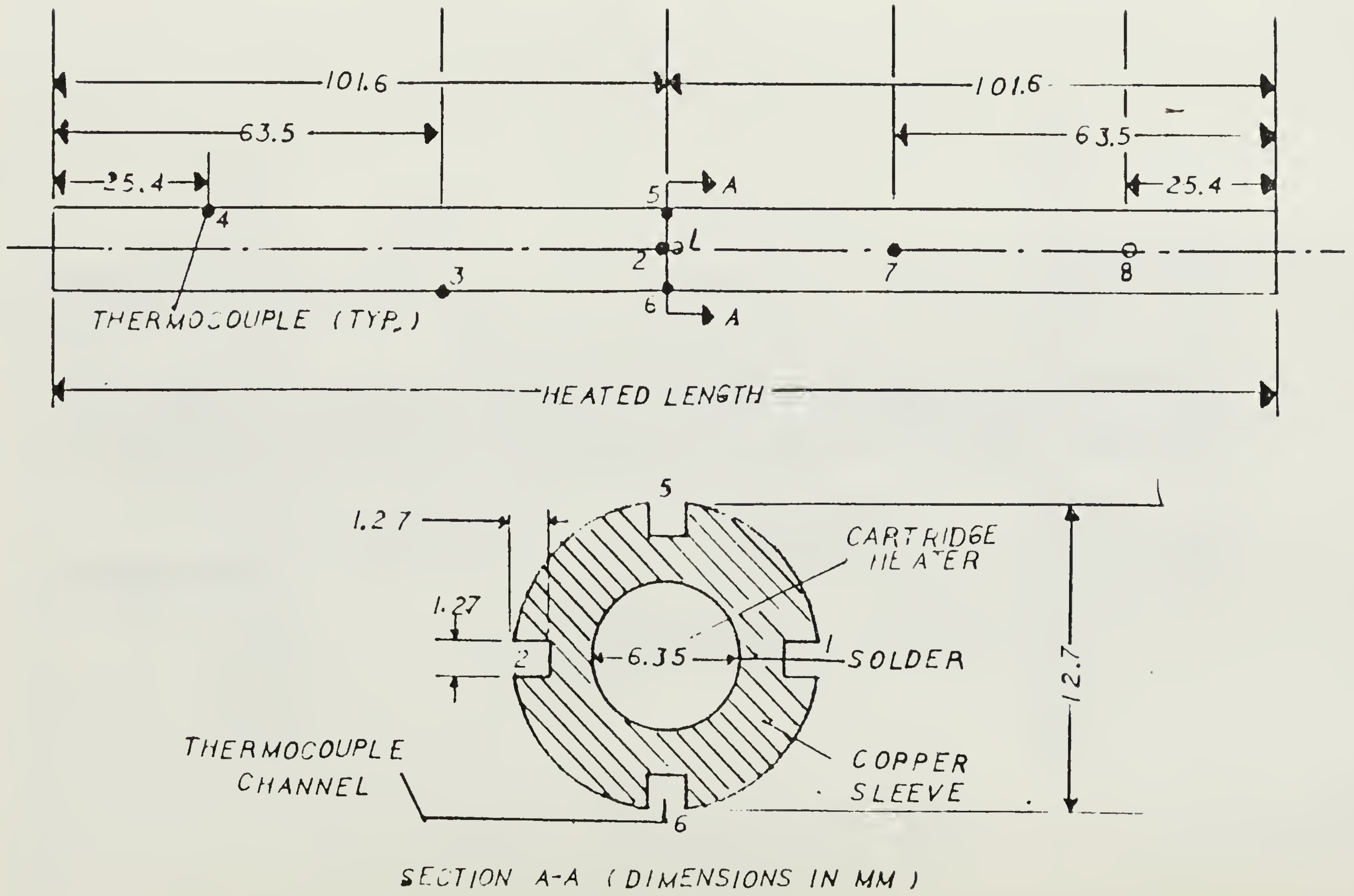


Figure 2.5 Positions of the Thermocouples.

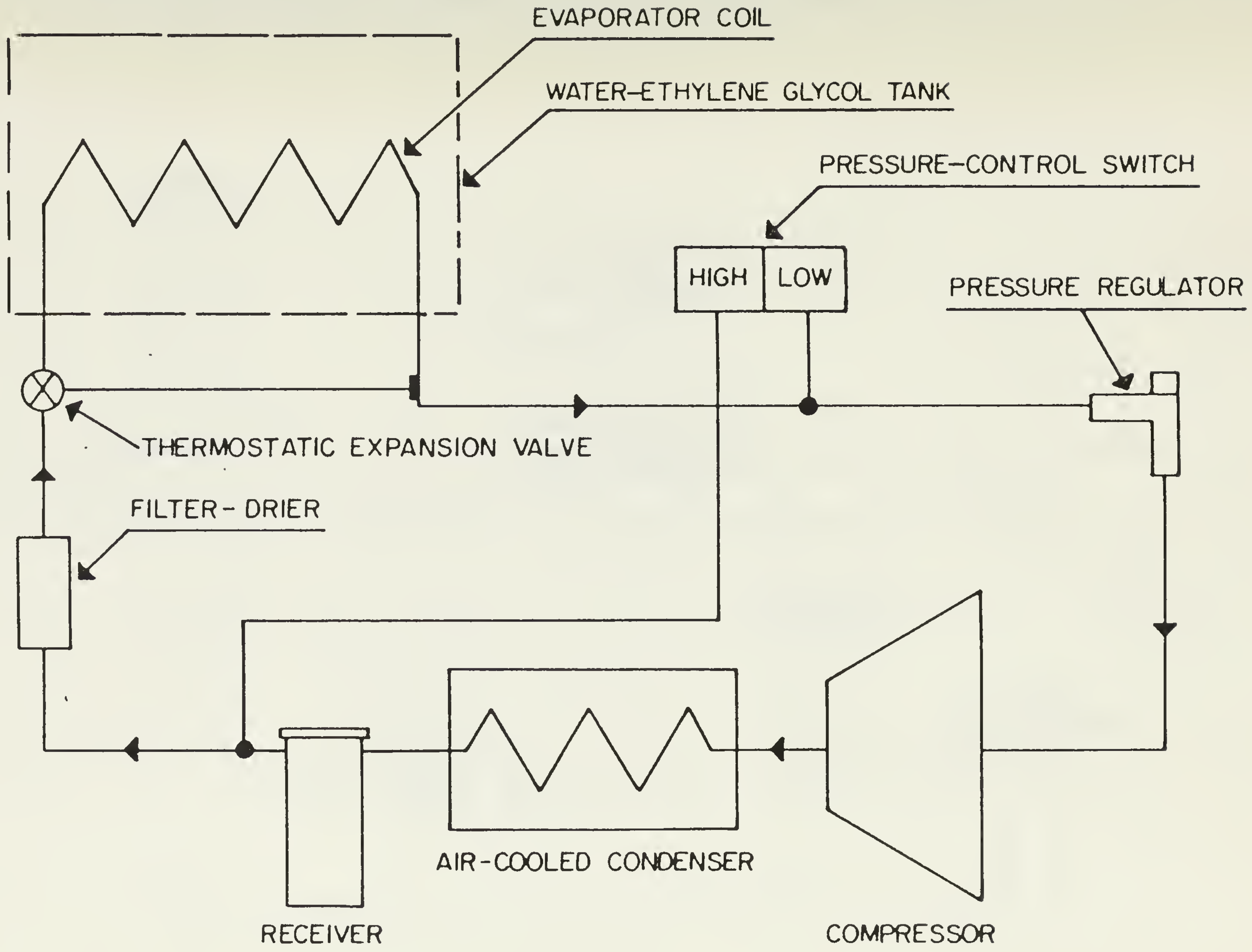


Figure 2.6 Schematic of R-12 Refrigeration Plant.

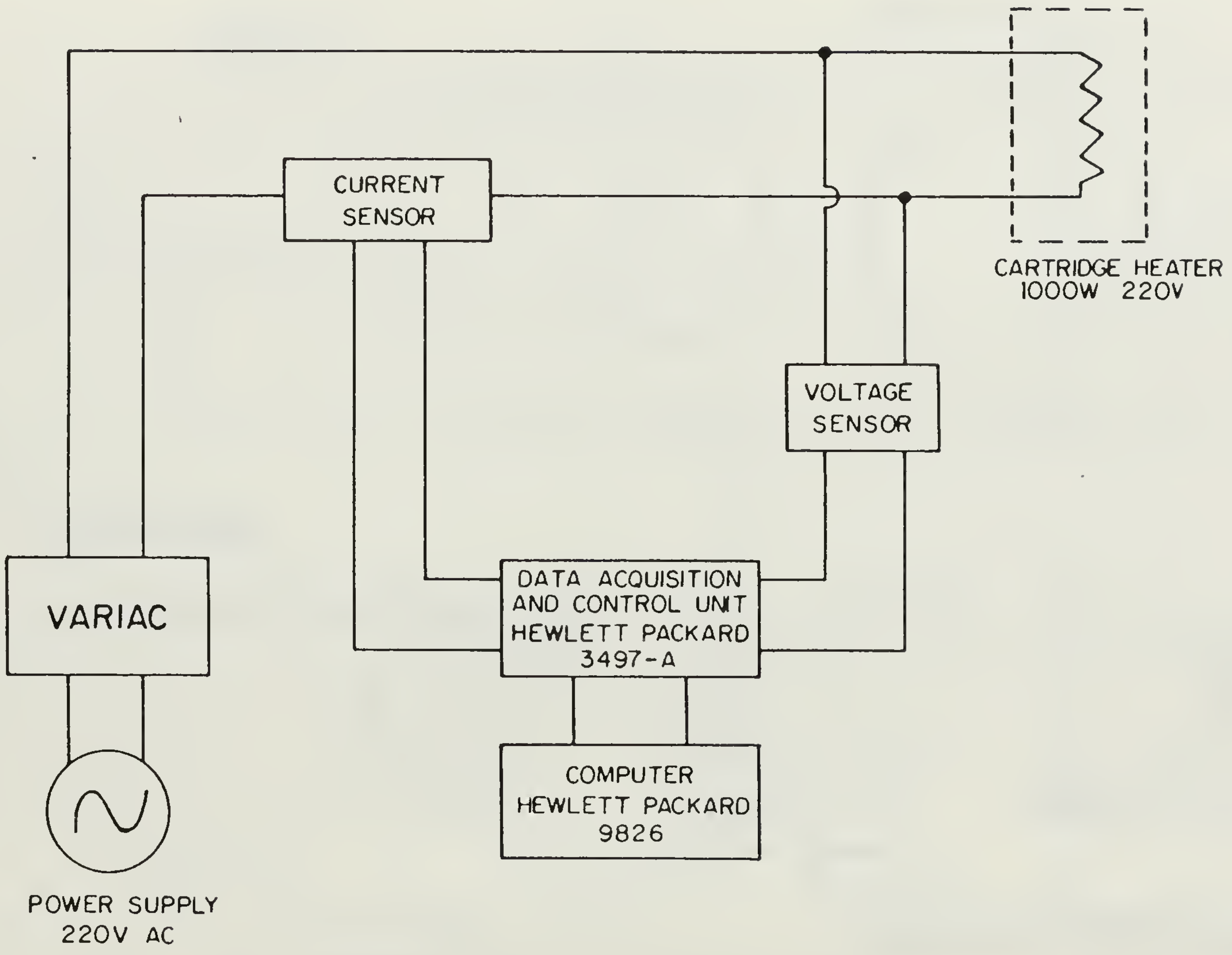

CURRENT SENSOR
CARTRIDGE HEATER 1000W 220V
VOLTAGE SENSOR
VARIAC
DATA ACQUISITION AND CONTROL UNIT HEWLETT PACKARD 3497-A
COMPUTER HEWLETT PACKARD 9826
POWER SUPPLY 220V AC

Figure 2.7 Schematic of the Power Measurement.

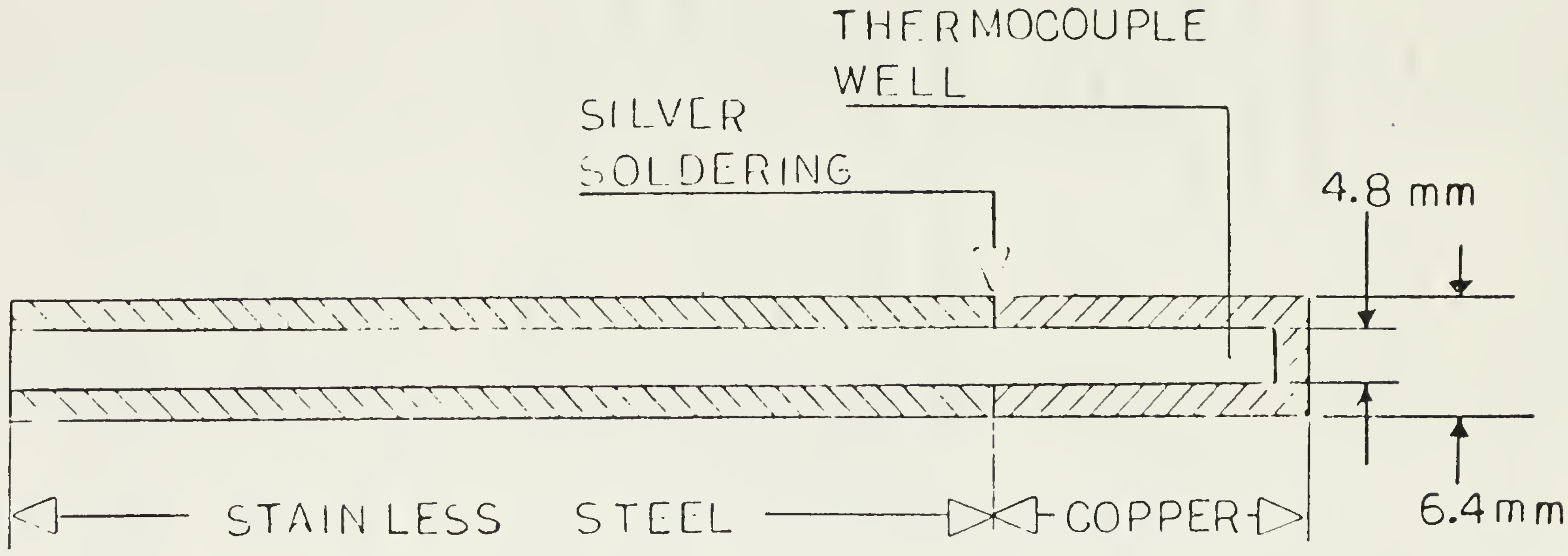


Figure 2.8 Sketch of a Thermocouple Well.

III. DATA ACQUISITION/REDUCTION

A. DATA ACQUISITION AND STORAGE

A Hewlett-Packard 3497A automatic data acquisition/control unit was used to read temperatures from the thermocouples and to read current and voltage values of the cartridge heater from the AC current sensor and the AC-DC true RMS converter, respectively. A Hewlett-Packard 9826A computer was used to control the data acquisition/control unit and to analyze and store data. Figure 3.1 shows a photograph of the data acquisition/control unit and computer.

Information was entered through the keyboard to prompt the data acquisition/control unit to automatically scan each channel. Channel assignments are listed in Table I. These raw data were immediately processed and a hard-copy printout was provided. Also, these data were transferred to a computer disk under a user-specified file name to keep a permanent record. The ability to store raw data directly enabled these data to be reduced at any time and allowed flexibility for changes to data-reduction software.

B. DATA REDUCTION

Following data acquisition for each data point, results were computed according to the stepwise procedure outlined in the next section, and then printed on a Hewlett-Packard 2671G thermal printer. Heat flux versus wall superheat (the temperature of the boiling surface minus the liquid temperature) were also stored in a user-specified plot data file for subsequent plotting using the subprogram PLOT. A Hewlett-Packard 7470A plotter was also interfaced with the computer.

C. STEPWISE DATA-COLLECTION AND SOLUTION PROCEDURE

1. Select tube type (all dimensions of the boiling test tube are included).
2. Set desired heat flux and saturation temperature of the boiling liquid. Wait for steady-state conditions (See Chapter 4 section B for details).
3. Scan all channels listed in Table I (thermocouples, AC current sensor and AC-DC true RMS converter readings).
4. Save these raw data in a user-specified file.
5. Convert these raw data readings to corresponding units (temperature in °C, current in Ampere, voltage in volt).
6. Compute the heat-transfer rate from the cartridge heater (See Chapter 4, Section C for details).
7. Compute the average wall temperature of the boiling test tube and calculate the wall superheat of this data set (See Chapter 4, Section C for details).
8. Compute the physical properties of R-114 using given correlations at film temperature (See Appendix H for details).
9. Compute the natural-convection heat-transfer coefficient of R-114 from non-boiling ends of the test tube.
10. Compute heat losses from non-boiling ends.
11. Calculate the heat flux from boiling test tube to the boiling liquid.
12. Calculate boiling heat-transfer coefficient of the R-114 from test tube.
13. Store the heat flux versus wall superheat values for each data set in a user-specified plot file.
14. Use the subprogram PLOT and plot the data run.

Appendix E shows a listing of computer program DRPR1 and subprogram PLOT and Appendix F shows an example of representative data run.

TABLE I

HP 3497A Channel Assignments

Channel	Assignment
25 - 32	Boiling tube wall temperature
33	Liquid temperature
34 - 35	Vapor temperature
36	Temperature of cooling liquid
62	AC-DC true RSM converter
63	AC current sensor

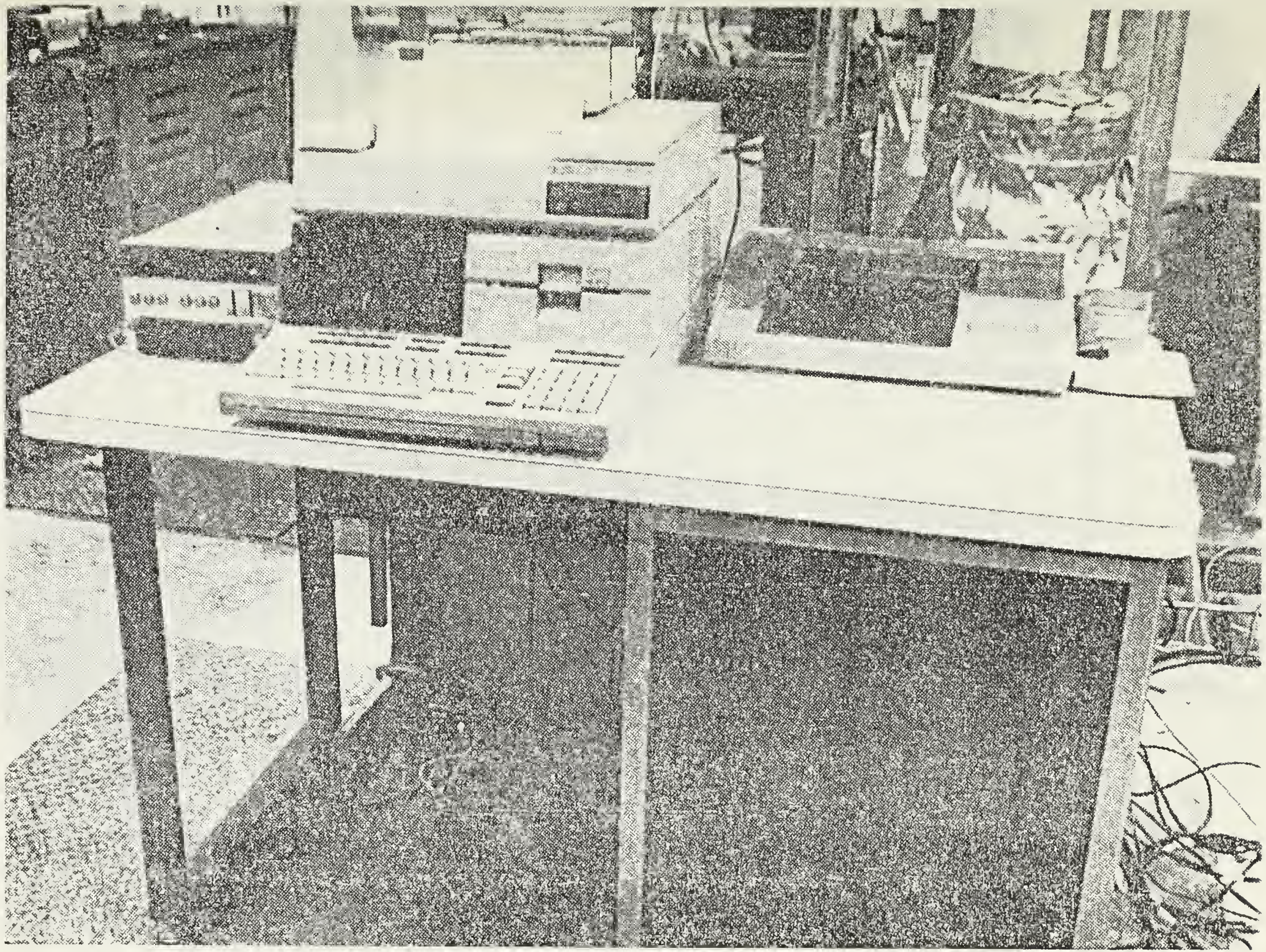


Figure 3.1 Photograph of Data Acquisition/Reduction System.

IV. EXPERIMENTAL PROCEDURE

A. PREPARATION

1. Pressure Test of the Apparatus with Air

Upon assembling all the components, including the boiling test tube, the experimental apparatus was pressurized with air up to 100 kN/m² (15 psi). A soap-bubble test was first carried out to locate any leaks from each component of the apparatus. All detected leaks were succesfully and systematically fixed at the end of each pressure test.

2. Pressure Test of the Apparatus with R-114

A second pressure test was performed using R-114 vapor. The apparatus were charged with R-114 vapor in the following manner. The apparatus was evacuated to 25 in. Hg. by a portable mechanical vacuum pump through valves V11 and V8 (See Figure 2.1 for the configuration of valves). Upon closing valve V11, a vapor outlet of the R-114 supply cylinder was connected to the apparatus by means of valve V10. Thus, the pressure of the apparatus was increased up to the saturation pressure of R-114 at that ambient temperature (ambient temperature was 21 °C and the corresponding saturation pressure (gage) was 86 kN/m² (12.5 psi)). An Automatic Halogen Leak Detector, TIF 5000, was used to detect R-114 leakage from the apparatus. The sensitivity of this detector is 1/2 oz. per year, 3 PPM max. A few small leaks were observed and successfully isolated.

3. Charging the Apparatus with R-114

Upon completing the pressure tests, the evaporator of the apparatus was filled with R-114 liquid up to a marked position (20 mm above the boiling test tube) with the following procedure (See also Figure 2.1).

1. The temperature of the water-ethylene glycol tank was reduced to about -17 °C by the R-12 refrigeration unit.
2. The apparatus was evacuated to about 29 in Hg vacuum by the vacuum pump (the valves V11 and V8 were open and V10 was closed).
3. Upon closing V11, the vapor outlet of the R-114 supply cylinder was connected to V10.
4. The cooling section pump 8, delivered the cooling liquid from the water-ethylene glycol tank to the condenser through the control valve VC.
5. The R-114 vapor condensed on the condenser coils and the liquid was collected in the evaporator by gravity, while V5 was open and V6 was closed.
6. Pump 8 was then stopped when the desired liquid level of the evaporator was achieved.
7. The pressure of the apparatus was then allowed to increase up to the saturation pressure corresponding to the ambient temperature.

B. NORMAL OPERATION

The following procedure was established to obtain the heat-transfer coefficient of R-114 from the smooth copper test tube:

1. The test tube was immersed in the pool (20 mm below the liquid level).
2. The R-12 refrigeration unit was operated 24 hours in advance in order to reduce the temperature of the water-ethylene glycol tank to about -17 °C.
3. The data acquisition/control unit, computer, power supply and cooling section pump were switched on.

4. Any accumulated noncondensible gases were evacuated by the portable mechanical vacuum pump through valve V8 and V11 (see Figure 2.1).
5. The iteractive computer program DRPR1 was operated and the desired heat flux and saturation temperature were given to the program as reference values.
6. The cartridge-heater voltage was then adjusted by variac in order to maintain the desired heat flux. The desired and actual heat fluxes were compared continiously by the computer program until they agreed to within 2 percent.
7. The actual versus desired saturation temperature of the liquid R-114 was also monitored by computer program. The amount of cooling liquid, which was being circulated through the copper coil in the condenser, was regulated by control valve VC in order to obtain nearly constant saturation temperature (±0.2 °C) at a given heat flux.
8. For all the consecutive settings during both increasing and decreasing heat fluxes, the boiling was allowed to stabilize for five minutes at each power setting and raw data (thermocouple readings, AC current sensor readings and AC-DC true RMS converter readings) were recorded in a user-specified file.
9. At a given power setting and a saturation temperature, the following processed data were recorded as a printout: wall temperatures of the boiling section, liquid bulk temperature, vapor temperature, net heat flux, temperature of the water-ethylene glycol mixture, wall superheat and heat-transfer coefficient of the R-114.
10. For each data set, the above procedure beginning with step 5 was repeated.

11. For each data series, seven different heat fluxes (1, 2, 5, 10, 20, 50 and 95 kW/m²) were selected.

C. HEAT-FLUX CALCULATION

According to the description of the boiling tube construction, the cartridge heater, which was inserted into the boiling tube is the heat source. The variations in heat flux are made through different voltage settings of the variac.

$$Q_H = VI \tag{1}$$

where:

Q_H = heat-transfer rate from the cartridge heater (W)

V = Voltage across the cartridge heater element (volts)

I = current through the heater element (amps)

This circuit is connected in series with an AC current sensor and in parallel with an AC-DC true RMS converter. Each of these two sensors produces a DC output in the range of 0-10 V, which is automatically read by the data acquisition/control unit. The calibration equations of these sensors give the actual values as follows:

$$V = 25V_S \tag{2}$$

$$I = I_S \tag{3}$$

where:

V_s = voltage reading by AC-DC true RMS converter (volts)

I_s = current reading by AC current sensor (volts)

The total length of the boiling test tube was 355.6 mm, while the active boiling length; i.e., the length of the cartridge heater, was 203.2 mm. The geometry of the boiling test tube is shown schematically in Figure 4.1.

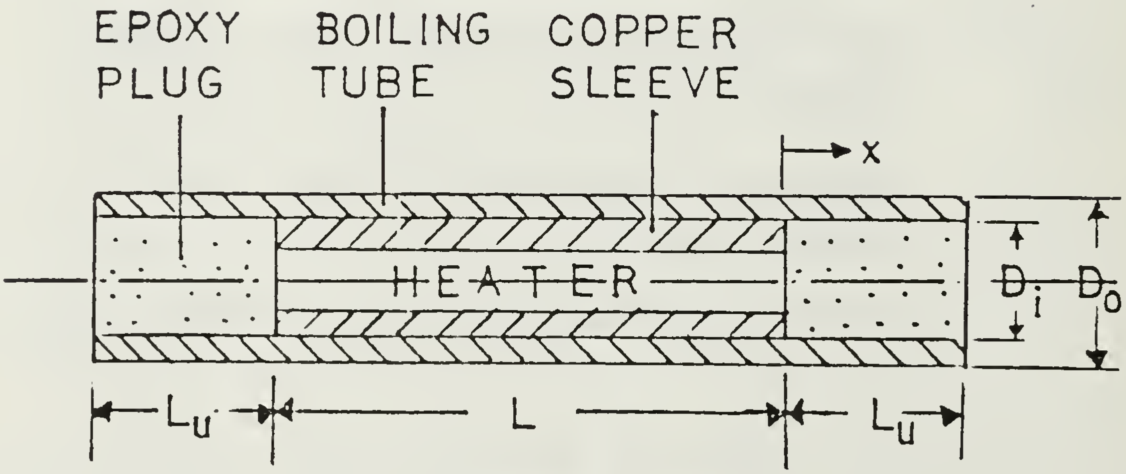


Figure 4.1 Geometry of the Boiling Test Tube.

Natural-convection heat transfer was assumed to occur at both ends of the test tube (the distance L_u). This part of the tube was considered to be a straight fin of uniform cross section. For this cylindrical geometry:

$$A_c = \pi\,(D_o^2 - D_i^2)/4 \qquad (4)$$

$$A_s = p\,L_u \qquad (5)$$

where:

A_c = cross-sectional area of tube (m^2)

D_o = tube outside diameter (m^2)

D_i = tube inside diameter (m^2)

A_s = tube outside surface area of nonboiling section (m^2)

p = tube outside wall perimeter (m)

p = πD_o

L_u = non-boiling length of the test tube (m)

It was assumed that the temperature at the base of the straight fin was equal to the average wall temperature of the active boiling section. The average outer wall temperature of the active boiling section was calculated using the radial conduction equation from the thermocouple position to the surface of the tube.

$$T_{avg} = \Big(\sum_{n=1}^{8} T_n\Big)/8 \qquad (6)$$

$$\bar{T}_{wo} = T_{avg} - Q_H\,(\,\ln(D_2 / D_1)/(2\,\pi\,L\,k_c)\,) \qquad (7)$$

$$T_b = \bar{T}_{wo}$$

where:

T_n = temperature of the thermocouple

location (°C)

T_{avg} = average wall temperature at the thermocouple locations (°C)

D_1 = diameter at the position of the thermocouple (m)

D_2 = outer diameter of the boiling tube (m)

L = active boiling tube length (m)

k_c = thermal conductivitiy of the copper (W/m.K)

$\bar{T}_{wo}$ = outer wall temperature of the boiling test tube (°C)

T_b = temperature at the base of straight fin (°C)

The temperature distribution of a fin of uniform cross-sectional area is given by [Ref. 20] with the following assumptions:

1. One-dimensional conduction in the x direction.
2. Steady-state condition.
3. Constant thermal conductivity.
4. Negligible radiation from the surface of the fin.
5. No heat generation in the fin.
6. Uniform convective heat-transfer coefficient over the fin surface.

$$d^2\theta / dx^2 - m^2\theta = 0 \qquad (8)$$

where:

$$\theta(x) = T(x) - T_{sat} \qquad (9)$$

and

$$m^2 = \bar{h}\, p \,/\, k_c A_c \qquad (10)$$

where:

T_{sat} = saturation temperature of fluid (°C)

$\bar{h}$ = convective heat-transfer coefficient ($W/m^2.K$)

Now, assuming negligible heat loss from the thin tube tip (at $x = L_u$)

$$(d\theta \,/\, dx)_{x=L_u} = 0 \qquad (11)$$

and also at the base:

$$Q_B = -k_c A_c (dT/dx)_{x=0} = -k_c A_c (d\theta/dx)_{x=0} \qquad (12)$$

where:

Q_B = heat-transfer rate from the base (W)

Using the above boundary conditions (Eq. 11, 12), the temperature distribution along the straight fin can be expressed as:

$$(\theta/\theta_b) = (\ \mathrm{Cosh}(m(L_u - x)) \,/\, \mathrm{Cosh}(mL_u)\ ) \qquad (13)$$

and the fin heat-transfer rate is:

$$Q_F = Q_B = \sqrt{\bar{h}\ p\ k_c A_c}\quad \theta_b \mathrm{Tanh}(mL_u) \qquad (14)$$

where:

Q_F = heat-transfer rate through one non-boiling length of the test tube (W)

The average difference between the wall temperature and the liquid temperature may now be determined from the following equation:

$$(\theta/\theta_b) = \left(\int_0^{L_u} (\mathrm{Cosh}(m(L_u - x))/\mathrm{Cosh}(mL_u))\ dx \right) / L_u \qquad (15)$$

$$\bar{\theta} = (\theta_b / mL_u)\ \mathrm{Tanh}(mL_u)$$

$$\bar{\theta} = \bar{T}_{wo} - T_{sat}$$

where:

$\bar{T}_{wo} - T_{sat}$ = average difference between outer wall temperature and liquid saturation temperature (K)

A free convection correlation stated by Churchill and Chu [Ref. 20] was applied for the average Nusselt number on a horizontal cylinder:

$$Nu_{D_o} = \left\{ 0.60 + \frac{0.387\ \overline{Ra}_{D_o}^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}} \right\}^2 \qquad (16)$$

$$10^{-5} < \overline{Ra}_{D_o} < 10^{12}$$

The average Nusselt number is:

$$\overline{Nu}_{D_o} = \frac{\bar{h}\ D_o}{k} \tag{17}$$

where:

k = thermal conductivity of R-114 (W/m.K)

Solving for $\bar{h}$:

$$\bar{h} = \frac{k}{D_o}\left\{0.60+\frac{0.387\ \overline{Ra}_{D_o}^{1/6}}{[1+(0.559/Pr)^{9/16}]^{8/27}}\right\}^2 \tag{18}$$

$$Pr = \frac{\nu}{\alpha} \tag{19}$$

$$\overline{Ra}_{D_o} = \frac{g\ \beta\ (\bar{T}_{W_O} - T_{SAT})\ D_o^3}{\nu\ \alpha} \tag{20}$$

$$\beta = -\ \frac{1}{\rho}\ \frac{\Delta\rho}{\Delta T} \tag{21}$$

where:

Pr = Prandtl number

ν = kinematic viscosity (m^2/s)

α = thermal diffusivity (m^2/s)

$\overline{Ra}_{Do}$ = average Rayleigh number

g = gravitational acceleration (m/s²)

β = volumetric thermal expansion coefficient (1/K).

ρ = density of R-114 liquid (kg/m³)

Now, substitution of Equations 10, 15, 20 into Equation 18 results in:

$$\bar{h} = \frac{k}{D_o}\left\{0.60 + 0.387\,\frac{\left[\dfrac{g\beta D_o^3 \theta_B \operatorname{Tanh}\left\{\left(\dfrac{\bar{h}P}{k_C A_C}\right)^{1/2} L_u\right\}}{\nu\alpha L_u\left(\dfrac{\bar{h}P}{k_C A_C}\right)^{1/2}}\right]^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}}\right\}^2 \qquad (22)$$

Equation 22 is solved for $\bar{h}$ by an iterative technique within a range of precision of 0.001. Knowing the value of the natural convection heat-transfer coefficient along the non-boiling ends, the total heat-loss rate is calculated from Equation 14, and the heat-transfer rate through the boiling section is obtained by subtracting the total heat-loss rate from the total heat-transfer rate:

$$Q_{Loss} = 2xQ_F \qquad (23)$$

$$Q = Q_H - 2xQ_F \qquad (24)$$

Finally, the heat flux from the boiling surface is:

$$q = Q / A_b \tag{25}$$

where:

$$A_b = \pi D_o L$$

V. RESULTS AND DISCUSSION

A. OUTLINE OF THE DATA RUNS

Using the procedure outlined in Chapter 3, nine data runs were completed, primarily to de-bug the experimental apparatus for its successful operation. For all data runs, the same smooth copper tube, described in Chapter 2, was used as the boiling surface. Each data run consisted of seven different heat fluxes (1000, 2000, 5000, 10000, 20000, 50000 and 95000 W/m^2) with a specified saturation temperature (0, 10 or 20 °C). All the raw data were stored in a user-specified file, named "WHxx", where "W" indicates Wieland tube, "H" indicates hard copper and "xx" indicates the data run number. Also, the heat-transfer coefficient versus heat flux data were stored in a plot file named "Pxx". A summary of these data runs is given in Table II.

B. LONGITUDINAL AND CIRCUMFERENTIAL TEMPERATURE VARIATIONS

During all data runs, especially at high heat fluxes (greater than 10 kW/m^2), large temperature variations were observed along the active boiling length of the test tube. Figure 5.1 shows the measured wall temperatures at each of the thermocouple locations, while Figure 5.2 shows the temperature variations in the circumferential direction (at the axial mid point) at two different heat fluxes (2 and 92 kW/m^2). Despite the uniform heat flux provided by the heater, the temperature measurements showed considerable variations in both axial and circumferential directions. The reason for these temperature variations may be explained as follows: As explained in Chapter 2, during the manufacturing of the boiling test tube, the copper sleeve was inserted into the boiling test tube without any shrink-fit process. The slip-fit process (with a diameter clearance of about 0.01 mm) used in this study was believed to have caused considerable thermal contact resistance at the interface between the boiling tube and the copper sleeve. Also, it is quite possible that this contact resistance was not uniform along both the axial and circumferential directions. The circumferential temperature distribution may also be attributed to a circumferential variation in boiling heat-transfer coefficient. During the high heat flux boiling regime, it was observed that the rate of bubble formation at the top of the horizontal boiling tube was lower than at the rest of the boiling tube; i.e., the boiling heat-transfer coefficient at top of the boiling tube was relatively lower than the other locations. This poorer performance at the top of the tube may be attributed to the fact that this portion was not receiving sufficient amount of fresh liquid to replace the liquid that evaporated. The obstruction of the liquid flow could be caused by the growing vapor blanket from the bottom to the top of the tube. Based on this observation, it is expected that the temperature at top of the boiling tube could be greater than the other locations. As shown in Figure 5.2, location 5, which represents the top of the horizontal boiling tube, measured the highest temperature difference as expected from the above-mentioned observation.

However, positions 1 and 2 showed a sizable discrepancy at high heat flux, and this was due most likely to the non-uniform contact resistance. Also, a comparison of thermocouples 2, 7 and 8 show a similar discrepancy. It is clear therefore, that the shrink-fit process must be used in the future.

C. PLOT ANALYSIS OF NUCLEATE BOILING REGIME

Figure 5.3 shows a typical nucleate pool-boiling performance curve of the smooth copper tube in R-114. The observed behavior of this process is analyzed by studying the different regions of the boiling curve.

From point A to point B, a continuous increase in wall superheat ($\bar{T}_{wo} - T_{sat}$) is observed when heat flux is increased. Also, during the experimental runs, no bubbles were observed along the boiling tube in this region of the curve. This region corresponds to the natural-convection process.

From point B to point C, a reduction in wall superheat is observed, while the heat flux continuously increased. While the middle portion of boiling tube showed bubbles, the rest of the boiling tube showed no bubbles during the experiment in this region. This region is known as the mixed boiling region, where transition from natural convection to nucleate pool boiling heat transfer takes place. The wall superheat continued to decrease until numerous nucleation sites became active (point C).

After point C, the wall superheat starts to increase with increasing heat flux as shown by region C to D. A very high density of bubble formation was observed along the active boiling section of the test tube (only a few bubbles were observed on the nonboiling sections of the test tube). See Figure 5.4. This observation confirmed the validity of the natural convection model assumed for data reduction. However, the R-114 liquid returning to the evaporator from the condenser caused a mild turbulance in the entrance region of this return line in the evaporator. Inherently, this affects the natural-convective heat-transfer performance of nonboiling ends of the test tube.

When heat flux is gradually decreased, the curve follows a different path (from D to E) as shown in Figure 5.3. This is due to the existing, stable nucleation sites remaining active for a wide range of heat fluxes. Similar analysis can also explain Figure 5.5, which represents the same data run but on a different basis (heat-transfer coefficient versus heat flux).

D. REPRODUCIBILITY TEST OF THE APPARATUS

In order to test the reproducibility of the experimental apparatus, two runs were performed on two different days (data run number 5 and 7 in Table II) at the same saturation temperature. Figure 5.6 and Figure 5.7 show the comparison of these two data runs. It can be seen from these figures, that there is very good agreement between these two different runs. This agreement shows the ability of the apparatus to reproduce data runs revealing successful operation.

E. BOILING PERFORMANCE OF SMOOTH COPPER TUBE IN R-114

In order to test the validity of the data taken from the present experimental apparatus (despite the presence of errors owing to the thermal contact resistance mentioned earlier), an attempt was made to compare the present data with data found in the literature. For this purpose, data run number 5 (10 °C saturation temperature with decreasing heat-flux rate) was compared with two sources in the literature as discussed below.

1. Comparison with Chongrungreong-Sauer Correlation

A correlation proposed by Chongrungreong and Sauer [Ref. 7] for the nucleate boiling performance of refrigerants and refrigerant-oil mixtures is compared with run number 5. This correlation is based on a dimensional analysis and data from various sources: [Ref. 7], [Ref. 14], [Ref. 15], [Ref. 22] and [Ref. 23].

$$h = 0.05259 \left[\frac{(Q/A)D}{\mu_L h_{fg}}\right]^{0.569} \left[\frac{\mu_l C_L}{k_L}\right]^{0.395} \times [P]^{1.695} \left[\frac{D}{0.01588}\right]^{-0.444} \left[\phi_f \frac{\rho_l}{\rho_v}\right]^{1.579} \tag{26}$$

where:

h = heat-transfer coefficient ($W/m^2.K$)

Q = heat rate (W)

A = surface area (m^2)

D = characteristic dimension of heated surface (m^2)

μ_L = viscosity of saturated liquid (gr/m.s)

h_{fg} = latent heat of vaporization (W.s/g)

C_p = specific heat of liquid (J/g.K)

k_L = thermal conductivity of liquid (W/m.K)

P = pressure (atmospheres)

ϕ_f = volume fraction of pure refrigerant

ρ_L = liquid density (g/cm^3)

ρ_V = vapor density (g/cm^3)

Eq'n (26) agrees very well with theirdata [Ref. 7] for R-11 as well as with Stephan's data [Ref. 22] for oil-free refrigerants, R-11, R-12, R-13, R-21, R-22, R-113 and R-114 with errors smaller than 16 percent. Figure 5.8 presents a comparison of the current experimental data with eq'n (26). As can be seen from this figure, the current data are in excellent agreement with this predictive equation. (The analysis given in Appendix I shows that the uncertainty in the boiling heat-transfer coefficient for this experimental investigation is about 20%.) This agreement was somewhat unexpected since the present data includes some kind of contact resistance in it. If this contact resistance were known, and were removed from the data, a higher boiling coefficient would result.

2. Comparison with Data of Henrici

The nucleate boiling heat-transfer coefficient of R-114 and R-114-oil mixtures from a horizontal, smooth copper tube (30 mm in outer diameter) has been experimentally investigated by Henrici [Ref. 10]. Since Henrici's data were taken from a tube with a larger diameter than the current boiling tube diameter, his data were revised to include this difference. For this purpose, a semi-emprical correlation to show the effect of tube diameter, as developed by Cornwell et al. [Ref. 21], was used:

$$Nu = C\ Re^{2/3} \qquad (27)$$

where:

$$Re = (q\ D) / (h_{fg} \mu) \quad \text{and} \quad Nu = (h\ D) / k$$

$$C = 150 \text{ for refrigerant}$$

Using this relationship, Henrici's data were revised for a diameter of 15.875 mm (5/8 in.) used in this experiment.

$$h(D=15.875\ \text{mm}) = h(D=30\ \text{mm})\ (30/15.875)^{1/3}$$

$$h(D=15.875\ \text{mm}) = 1.24\ h(D=30\ \text{mm})$$

A line representing the revised Henrici's data is plotted in Figure 5.9 together with the unrevised data line. It can be seen that the boiling heat-transfer coefficient increases by about 25 percent when the diameter is decreased from 30 mm to 15.875 mm (50-percent reduction in diameter).

Figure 5.10 provides Henrici's natural convection data and Henrici's revised boiling data in comparison with the present experimental data. As expected, the natural-convective heat-transfer coefficient is lower than the boiling heat-transfer coefficient. The boiling heat-transfer coefficient from Henrici's data is higher than the coefficient found in this study. The reason for this may be explained as follows. In Henrici's experimental investigation, the measurement of the surface temperature has been done using thermocouples outside of the boiling surface. Unfortunately, he did not describe in detail how he measured the wall temperatures. If the thermocouples were not totally imbedded in the wall, they may produce incorrect wall temperature values. If a portion of a thermocouple is in direct contact with the liquid, this thermocouple will measure a value lower than the actual wall temperature and such an error would decrease the wall superheat, thus resulting in a higher heat-transfer coefficient. Also, if the contact resistance effect is removed from the present data, then the heat transfer coefficient would increase toward the Henrici data.

F. EFFECT OF PRESSURE

The effect of pressure on the nucleate-boiling heat-transfer performance of R-114 was investigated for reduced pressures (P/P_c) of 0.0268, 0.0391 and 0.0556 (data run numbers 5, 8 and 9, respectively in Table II). Figure 5.11 shows the relative increase in boiling heat-transfer coefficient with increase of pressure as expected. In order to calculate the pressure effect on the boiling performance, a simple equation developed by [Ref. 7] was used.

$$h = 6.17\ (q)^{0.55}\ (\phi)^{3.65}\ P^{0.24} \qquad (28)$$

where:

h = heat-transfer coefficient (W/m².K)
q = heat flux (W/m²)
ϕ = volume fraction of pure refrigerant
P = boiling pressure (atmospheres)

Using this equation, the pressure effect on the boiling heat transfer coefficient can be written as follows:

$$h(P_1) = h(P_2)\ (P_1 / P_2)^{0.24}$$

According to the above equation, the effect of pressure from 127 kN/m² to 87 kN/m² decreases the boiling heat-transfer coefficient by about ten percent, which agrees reasonably well with the current experimental data.

TABLE II

A Summary of the Data Runs

Data Run #	Pressure psia	Pressure kN/m^2	Liquid Temp (°F)	Liquid Temp (°C)	p/p_c	Heat Flux	Purpose
1	13.8	95.2	36	2.2	0.0292	Decreasing	Debugging
2	16.4	113.1	44	6.6	0.0347	"	"
3	11.9	82.1	28	-2.2	0.0251	"	"
4	18.5	127.7	50	10.0	0.0391	Increasing	Repeatability and Plot Analysis
5	18.5	127.7	50	10.0	0.0391	Decreasing	
6	18.5	127.7	50	10.0	0.0391	Increasing	"
7	18.5	127.7	50	10.0	0.0391	Decreasing	"
8	12.7	87.6	32	0.0	0.0268	Decreasing	Pressure Effect
9	26.3	181.4	68	20.0	0.0556	"	"

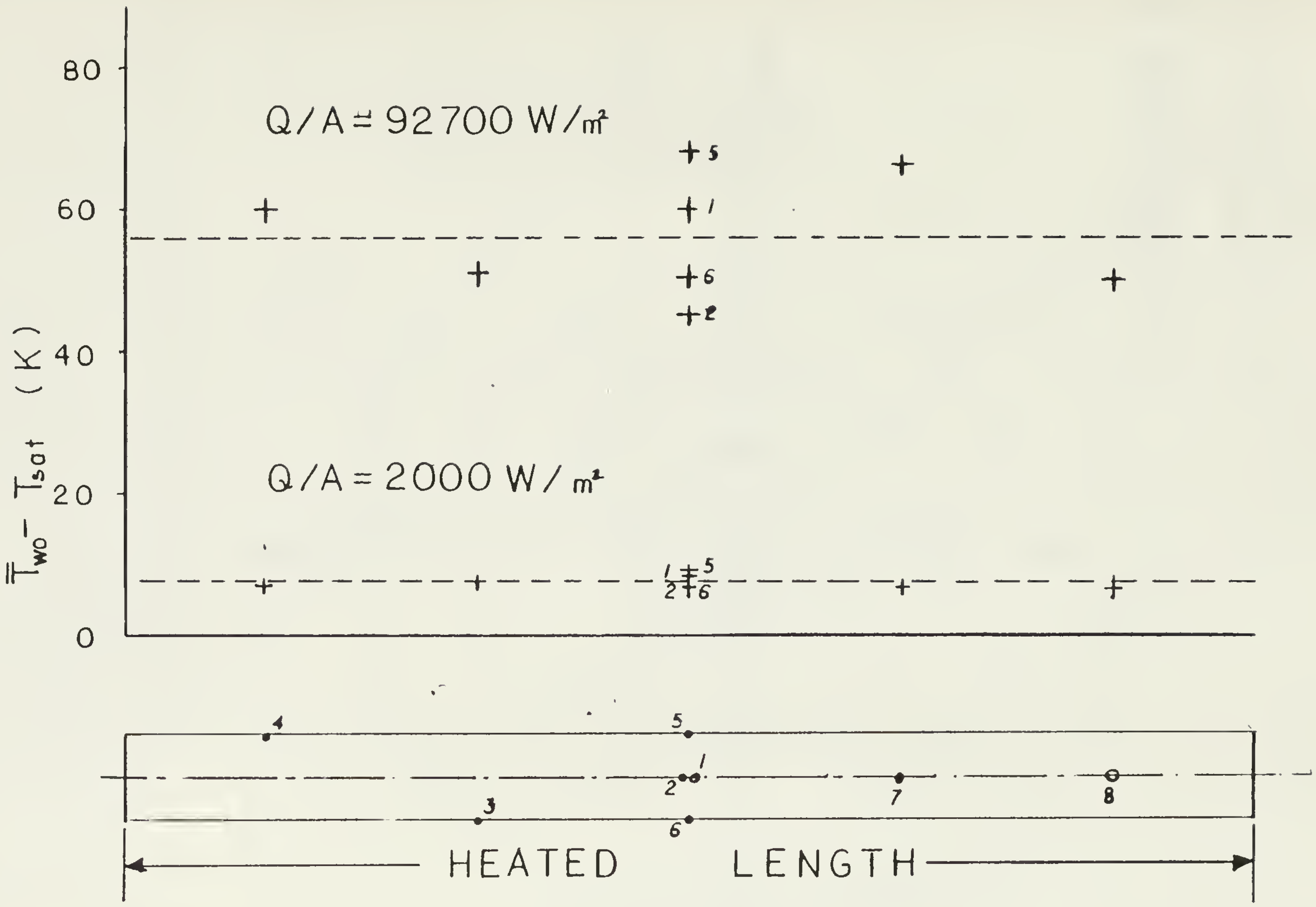


Figure 5.1 Temperature Variations on the Boiling Tube.

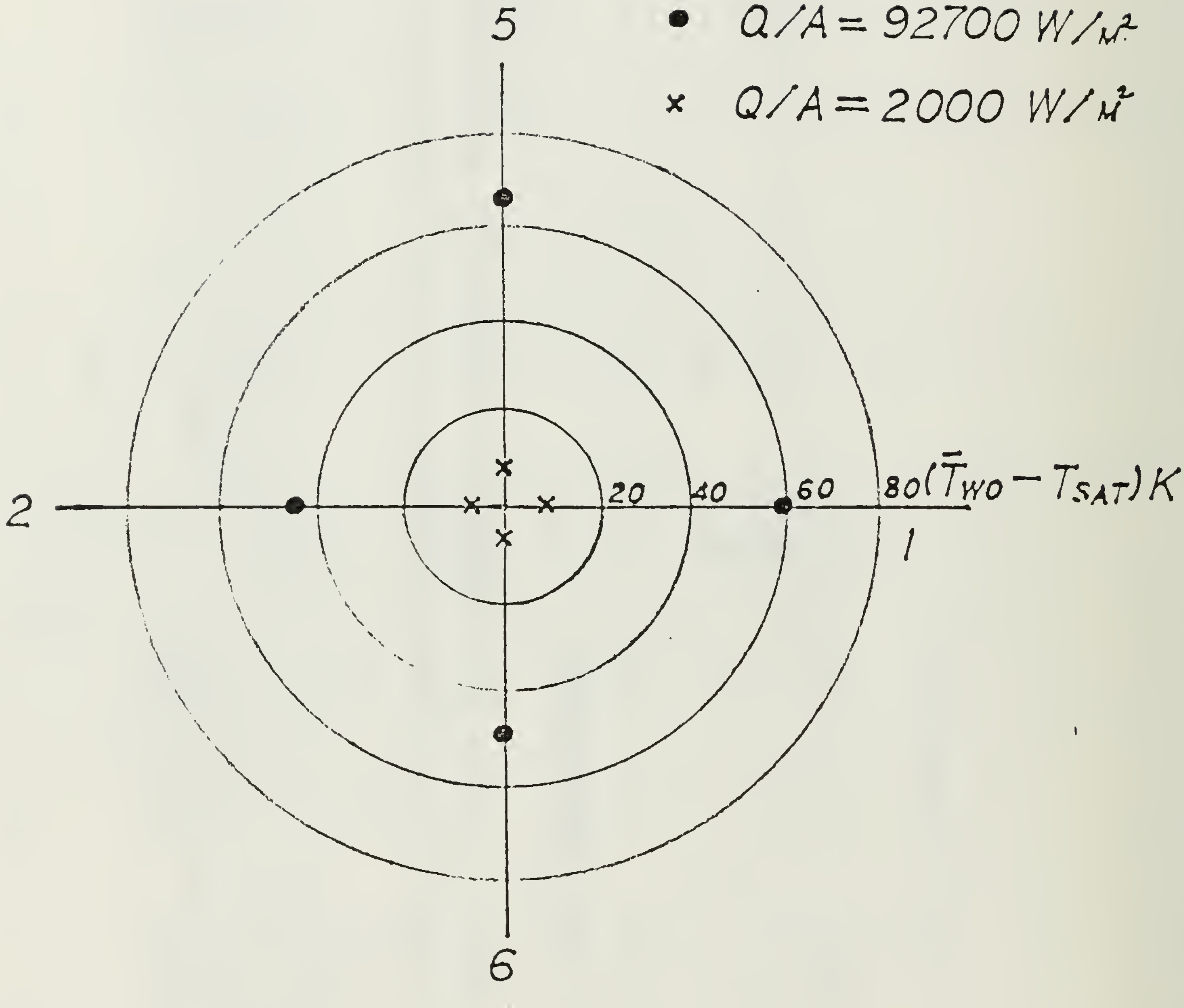


Figure 5.2 Circumferential Temperature Variation on the Boiling Tube.

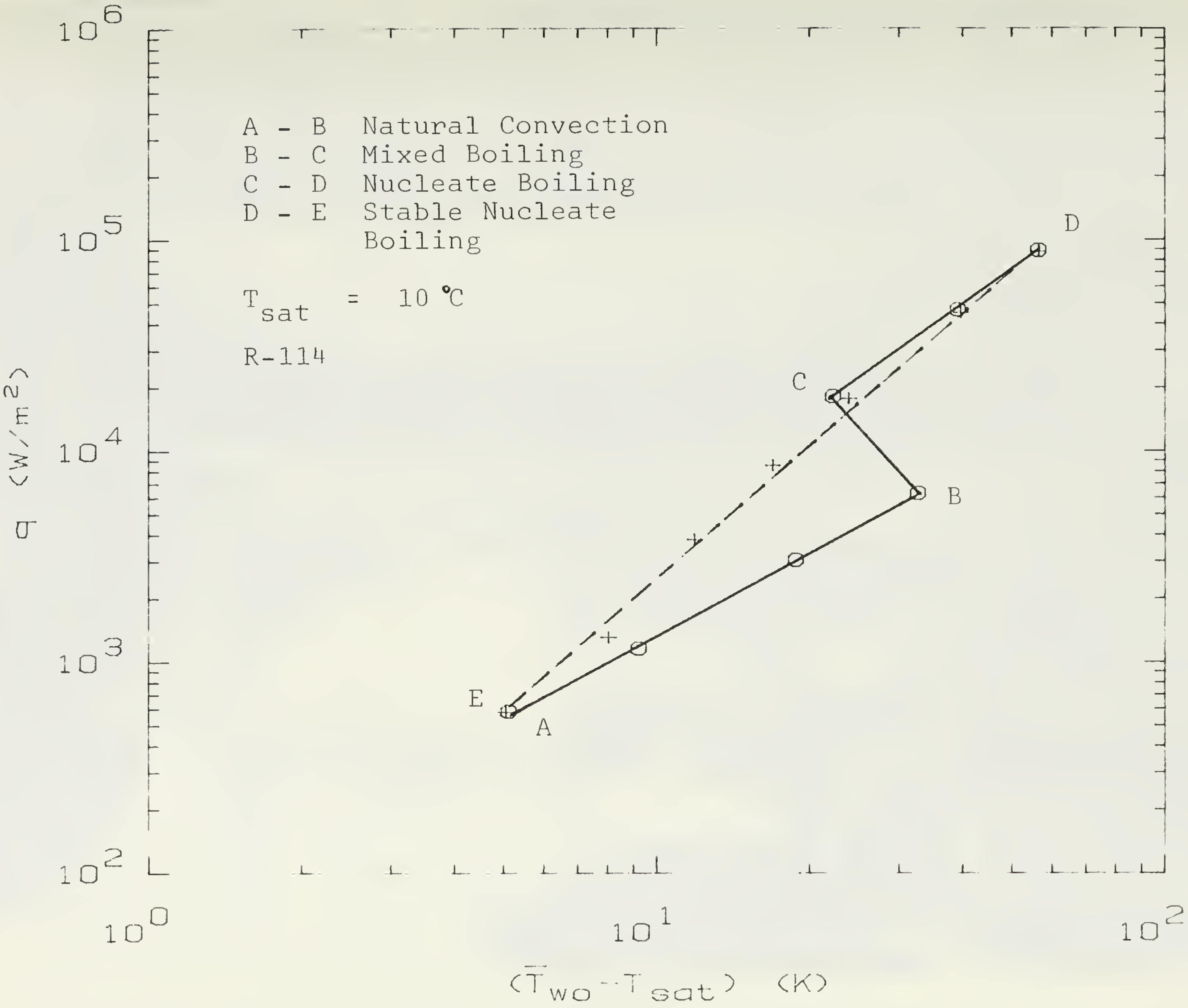


Figure 5.3 Typical Nucleate Pool Boiling Curve for R-114

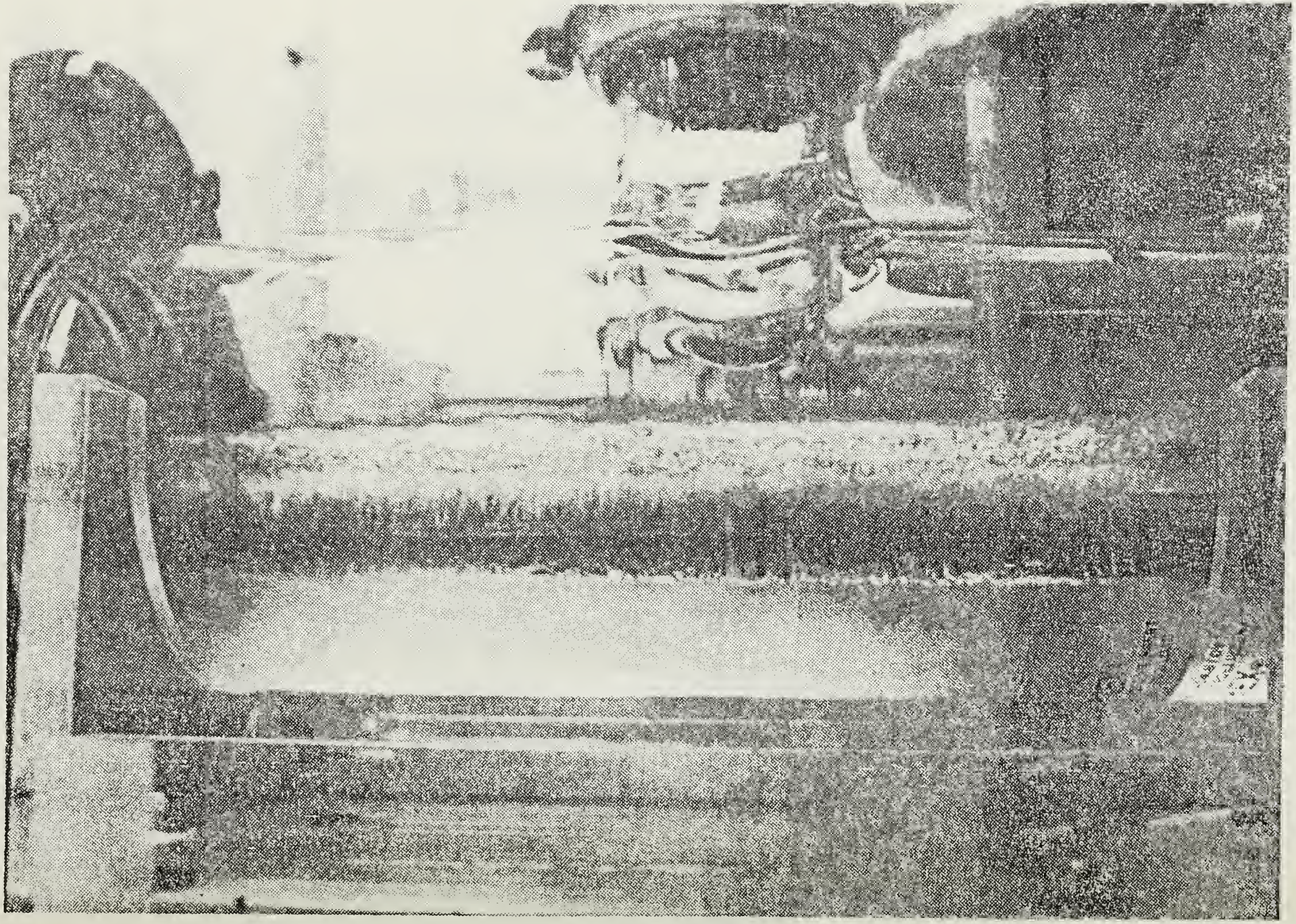


Figure 5.4 A Photograph of the Boiling Tube During High Heat Flux Operation.

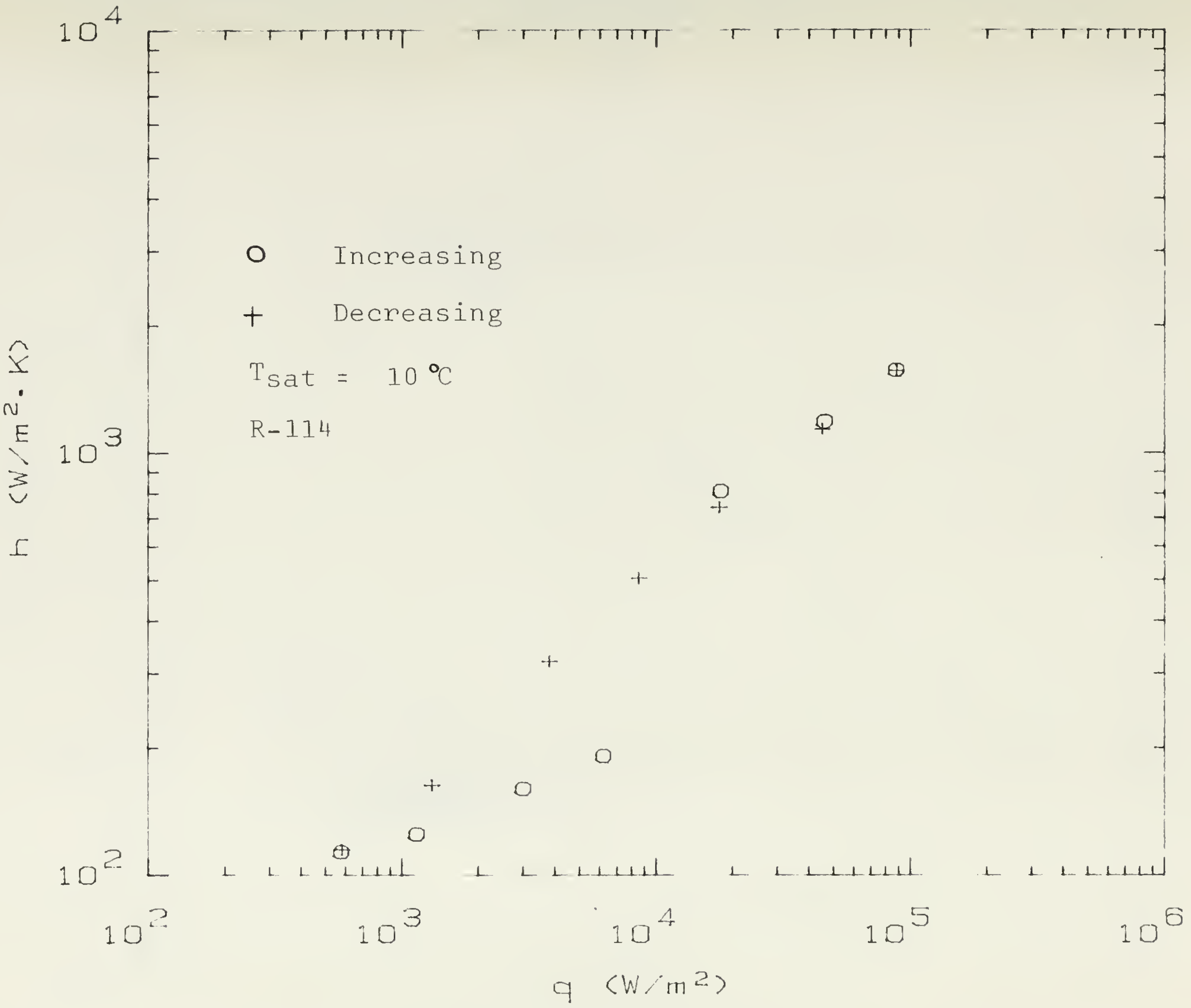


Figure 5.5 Typical Nucleate Pool Boiling Heat Transfer Coefficient Curve for R-114

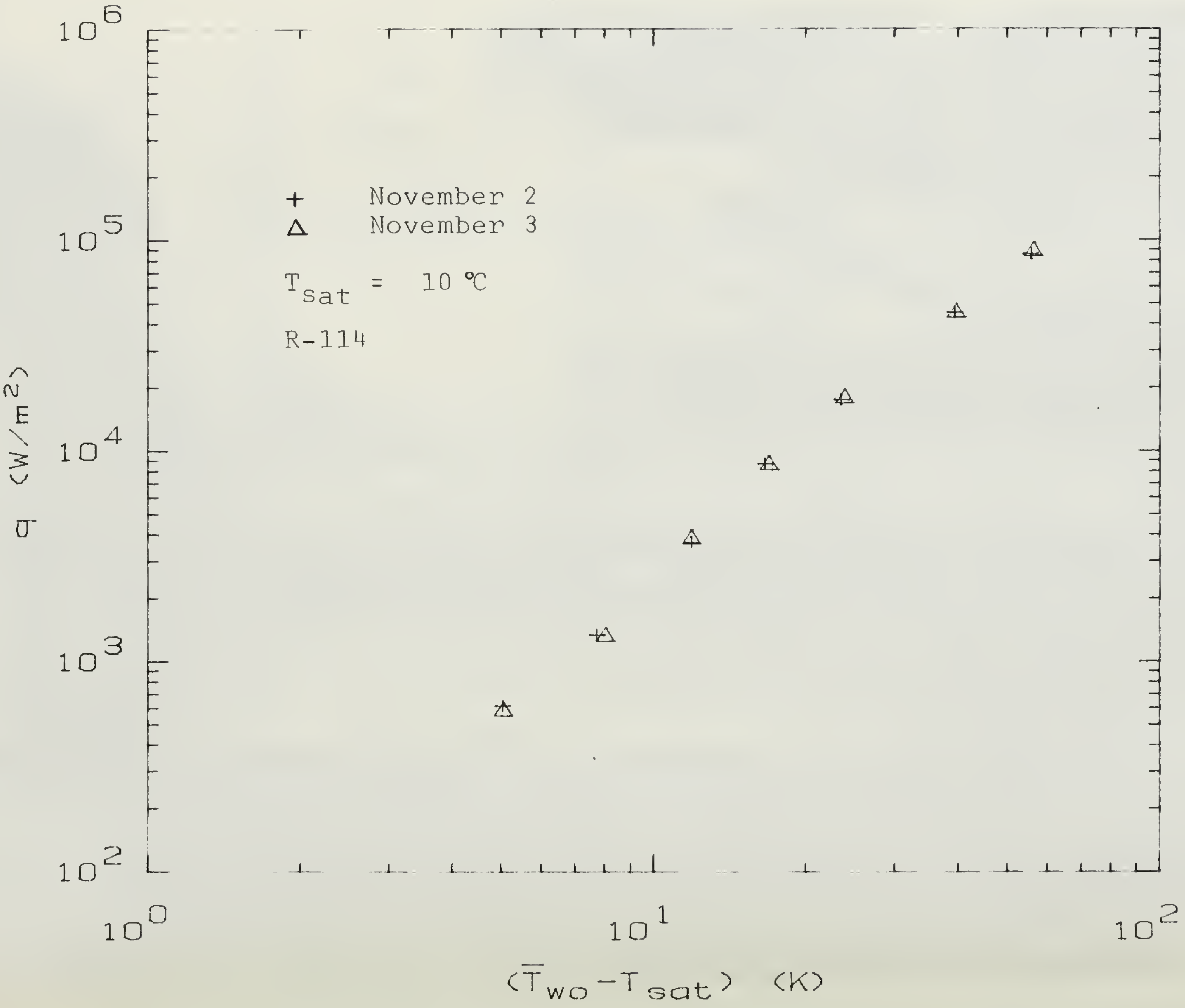


Figure 5.6 Reproducibility of the Apparatus.

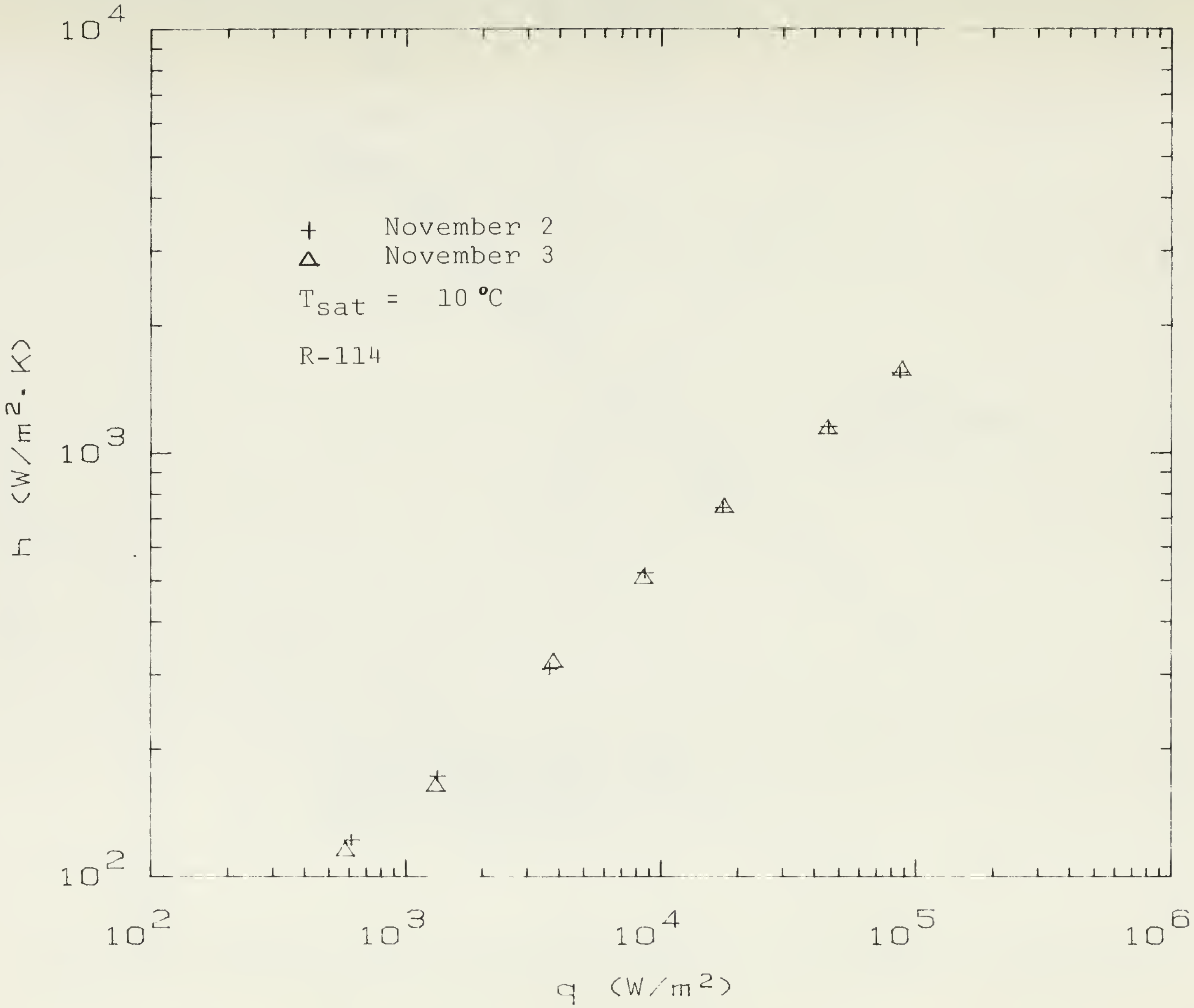


Figure 5.7 Reproducibility of the Heat-Transfer Coefficients.

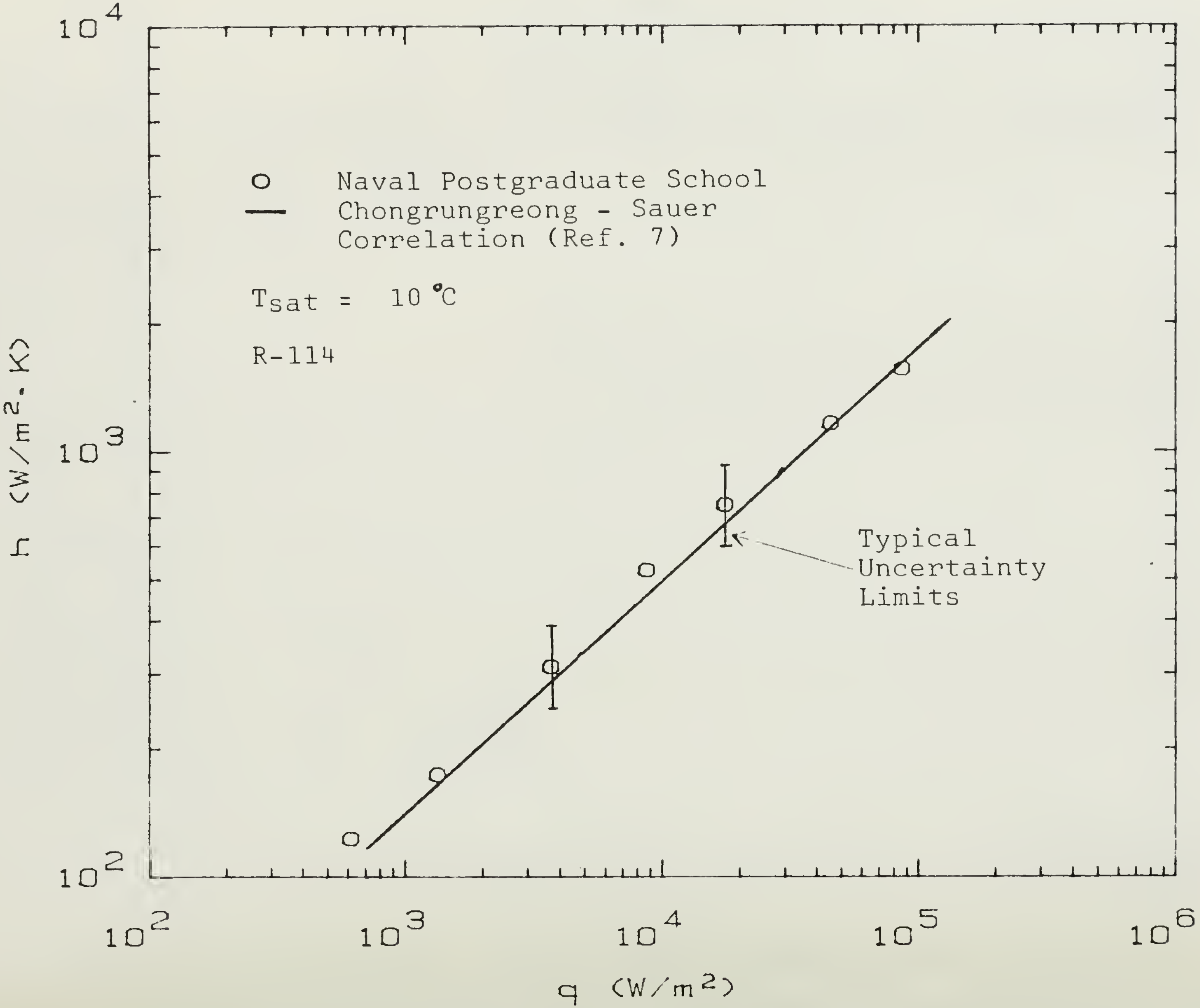


Figure 5.8 A comparison of the Boiling Performance with the Correlation of Chongrungreong and Sauer (Ref. 7).

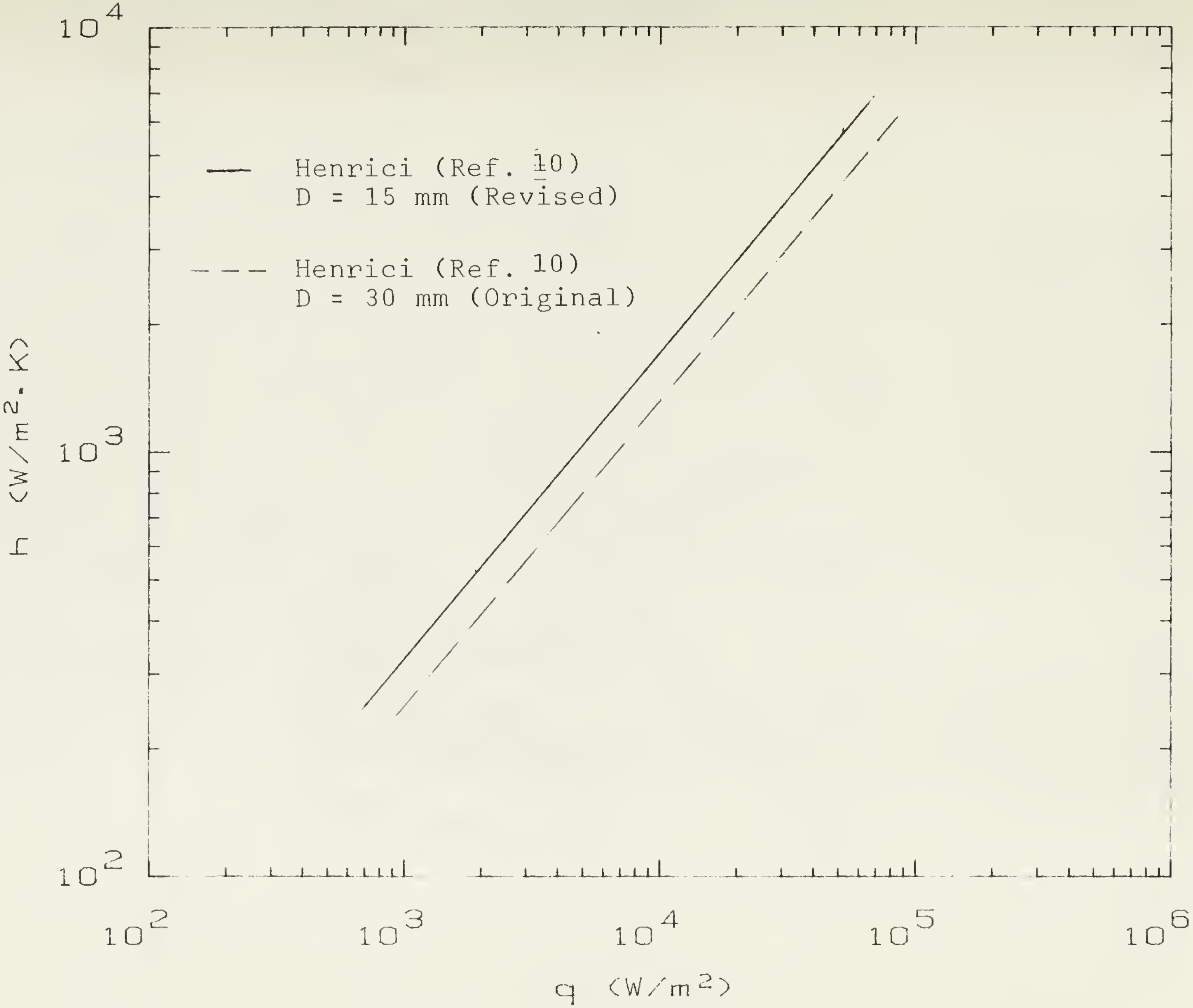


Figure 5.9 Effect of Tube Diameter on Boiling Performance.

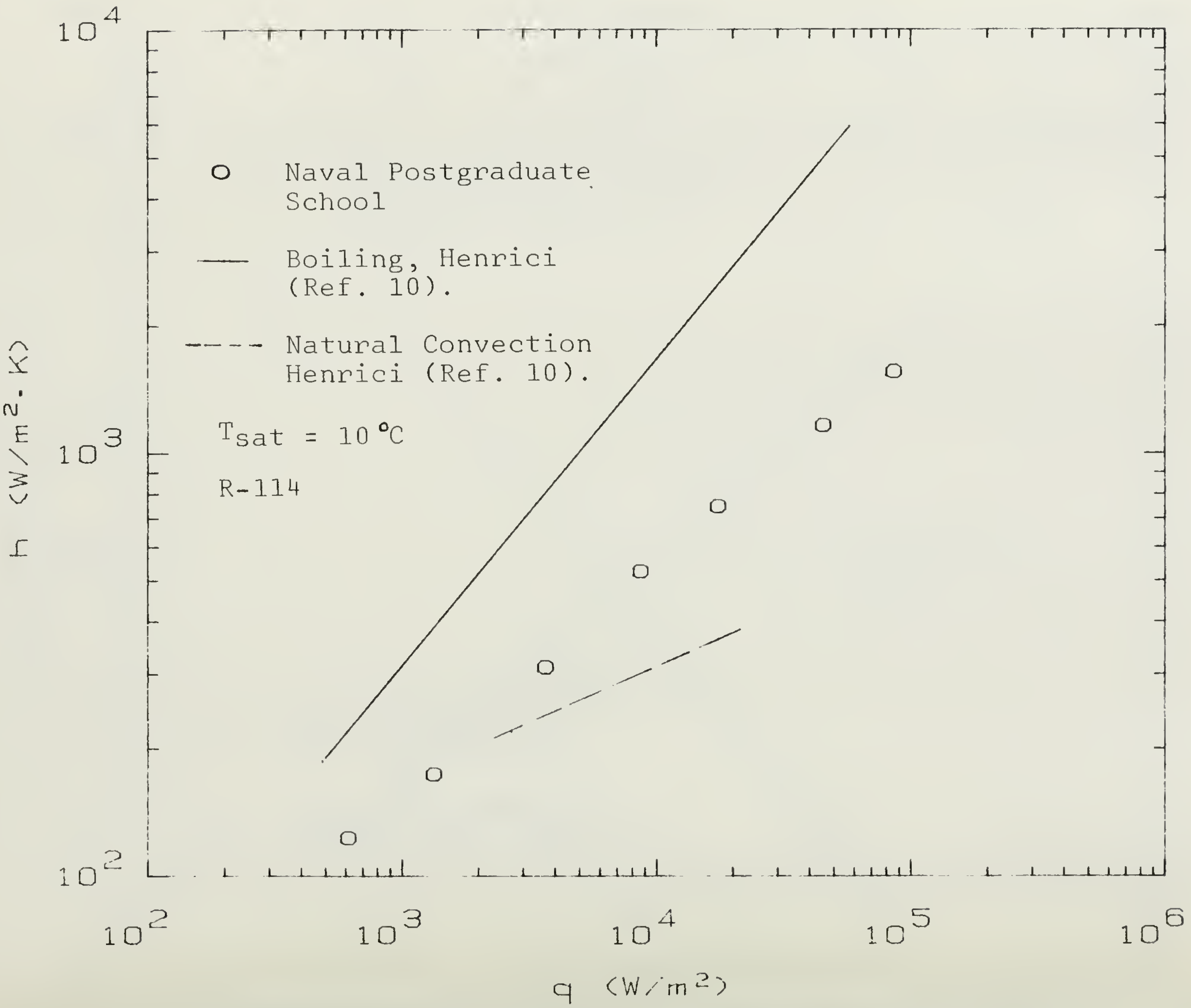


Figure 5.10 Comparison of Current Data with Data of Henrici (Ref. 10)

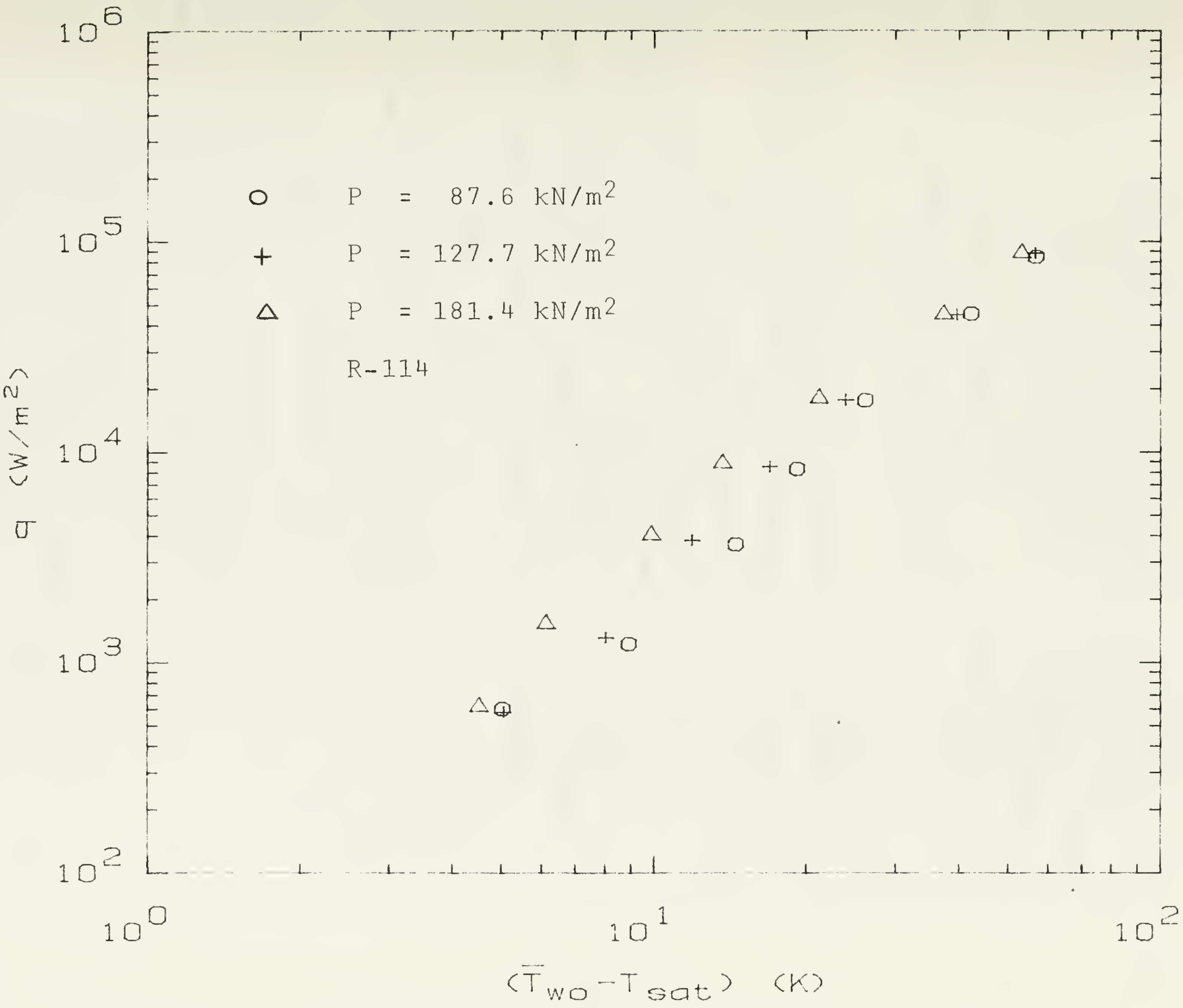


Figure 5.11 The Effect of Pressure on Boiling Performance.

VI. CONCLUSIONS

1. An experimental apparatus to study the nucleate-boiling heat-transfer performance of R-114 from a single horizontal tube has been designed, constructed and instrumented.
2. An analysis scheme to establish the nucleate boiling heat-transfer coefficient of R-114 has been outlined, and data acquisition/reduction programs have been written and tested.
3. The experimental apparatus was successfully operated in the saturation temperature range of 0 °C to 20 °C and in the heat flux range of 1 to 95 kW/m^2.
4. The test runs performed on different days on the apparatus showed repeatabilty within ± 1 percent, revealing successful operation of the apparatus.
5. The uncertainty in wall superhet (about 20%) drastically affected the boiling heat-transfer coefficient. This can be attributed to the large temperature variations measured along and around the boiling tube because of a non-uniform contact pressure between the copper sleeve and the boiling tube.
6. The data taken on this experimental apparatus have been compared with the experimental correlation for refrigerant and refrigerant-oil mixtures developed by Chongrungreong and Sauer [Ref. 7]. The present data agreed to within 5 percent with this correlation.
7. The current data were also compared with Henrici's [Ref. 10] experimental nucleate boiling data (from smooth horizontal copper tube in R-114). The present data lie as much as 50 percent below Henrici's data. This disagreement may be attributable to the thermal

contact resistance present in the boiling tube, or to the errors in Henrici's wall temperature measurements.

VII. RECOMMENDATIONS

1. To prevent considerable temperature variations along and around the boiling tube, a shrink-fit process must be applied in order to have more uniform contact pressure between the copper sleeve and the boiling tube.
2. To determine the effect of contact resistance (between copper sleeve and boiling tube) on the nucleate-boiling heat-transfer performance of a horizontal tube, a series of experimental investigations should be performed with different contact pressures.
3. To maintain constant electrical power input into the boiling tube, a voltage regulator should be added to the input line.
4. To keep the temperature of the water-ethylene glycol mixture at a constant value, the 1/2-ton capacity R-12 refrigeration unit should be replaced with a 2-ton unit.
5. The positive-displacement pump should be replaced with a centrifugal-type pump, in order to deliver lower flow rates without reaching an electrical overload limit.
6. To check the saturation temperature of the boiling liquid corresponding to the measured saturation pressure, a more accurate pressure transducer should be used.
7. The active length of the cartridge heater should be carefully measured or obtained from the manufacturer.

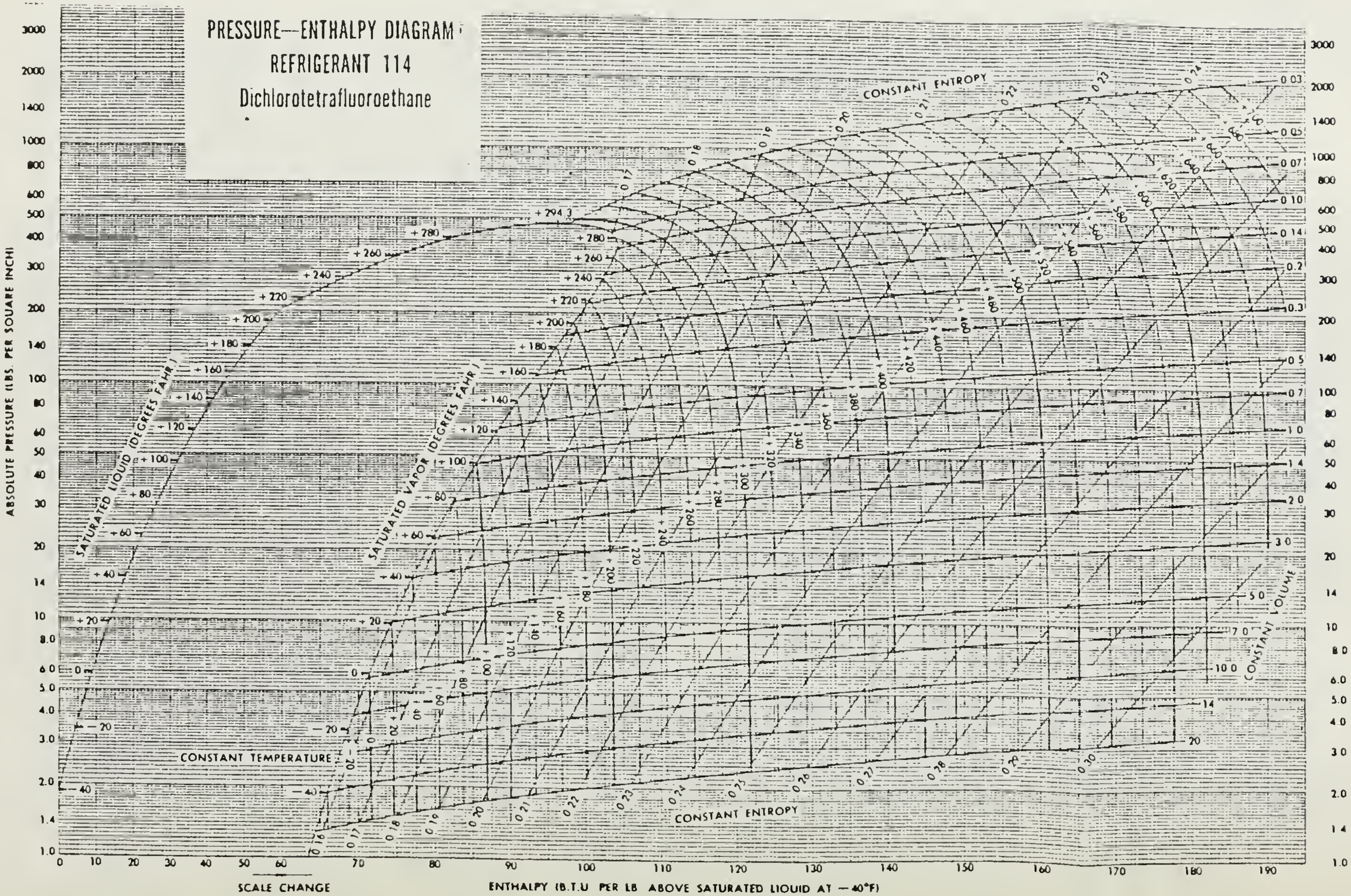
APPENDIX A

PHYSICAL PROPERTIES OF FREON FLUOROCARBON COMPOUNDS

		"FREON" 11	"FREON" 12	"FREON" 113	"FREON" 114
Chemical Formula		CCl_3F	CCl_2F_2	$CCl_2F\ CClF_2$	$CClF_2\ CClF_2$
Molecular Weight		137.37	120.92	187.38	170.93
Boiling Point at 1 atm	°C °F	23.82 74.87	−29.79 −21.62	47.57 117.63	3.77 38.78
Freezing Point	°C °F	−111 −168	−158 −252	−35 −31	−94 −137
Critical Temperature	°C °F	198.0 388.4	112.0 233.6	214.1 417.4	145.7 294.3
Critical Pressure	atm lbs/sq in abs	43.5 639.5	40.6 596.9	33.7 495	32.2 473.2
Critical volume	cc/mol cu ft/lb	247 0.0289	217 0.0287	325 0.0278	293 0.0275
Critical Density	g/cc lbs/cu ft	0.554 34.6	0.558 34.8	0.576 36.0	0.582 36.32
Density, Liquid at 25°C (77°F) at 54.5°C (130°F)	g/cc lbs/cu ft g/cc lbs/cu ft	1.476 92.14 1.40 87.40	1.311 81.84 1.19 74.28	1.565 97.69 1.49 93.01	1.456 90.91 1.36 84.90
Density, Sat'd Vapor at Boiling Point	g/l lbs/cu ft	5.86 0.367	6.33 0.395	7.38 0.461	7.83 0.489
Specific Heat, Liquid (Heat Capacity) at 25°C (77°F)	cal/(g) (°C) Btu/(lb) (°F)	0.208	0.232	0.218	0.243
Specific Heat, Vapor, at Const Pressure (1 atm) at 25°C (77°F)	cal/(g) (°C) Btu/(lb) (°F)	0.142 @ 38°C (100°F)	0.145	0.161 @ 60°C (140°F)	0.170
Specific Heat Ratio of Vapor at 25°C and 1 atm	Cp/Cv	1.137 @ 38°C (100°F)	1.137	1.080 @ 60°C (140°F)	1.084
Heat of Vaporization at Boiling Point	cal/g Btu/lb	43.10 77.51	39.47 71.04	35.07 63.12	32.51 58.53
Thermal Conductivity at 25°C (77°F) Btu/(hr) (ft) (°F) Liquid* Vapor (1 atm) (Data from ASHRAE in most cases)		0.0506 0.00451	0.0405 0.00576	0.0434 0.0014 (0.5 atm)	0.0372 0.0060
Viscosity at 25°C (77°F) Liquid Vapor (1 atm) (Data from ASHRAE in most cases)	centipoise centipoise	0.430 0.0107	0.214 0.0123	0.68 0.010 (0.1 atm)	0.36 0.0112
Surface Tension at 25°C (77°F) dynes/cm		18	9	17.3	12
Refractive Index of Liquid at 25°C (77°F)		1.374	1.287	1.354	1.288
Relative Dielectric Strength of Vapor at 1 atm and 25°C (77°F) (nitrogen = 1)		3.71	2.46	3.9 (0.44 atm)	3.34
Dielectric Constant Liquid Vapor (1 atm)*a		2.28 @ 29°C 1.0036 @ 24°C*b	2.13 1.0032 @ 29°C	2.41 @ 25°C	2.26 @ 25°C 1.0043 @ 26.8°C
Solubility of "Freon" in Water at 1 atm and 25°C (77°F)	wt %	0.11	0.028	0.017 (Sat'n Pres)	0.013
Solubility of Water in "Freon" at 25°C (77°F)	wt %	0.011	0.009	0.011	0.009
Solubility Parameter ( ) Kauri-Butanol Value (KB)		7.5 60	6.1 18	7.2 32	6.2 12
Toxicity* Threshold Limit Value (TLV)	ppm (v/v) mg/m³	1000 5600	1000 4950	1000 7600	1000 7000

APPENDIX B

PRESSURE-ENTHALPY DIAGRAM OF R-114



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APPENDIX C

APPLICATIONS OF "FREON" FLUOROCARBON COMPOUNDS

The table below is intended to provide a general view of the range of applications and is not all inclusive. For specialized applications or more detail, please make specific request.

Fluorocarbon	Refrigerants	Aerosol Propellants[b]	Solvents, Blowing Agents, Fire Extinguishants, Dielectric Fluids and Other Uses
"Freon" 14	—	—	—
"Freon" 23	Component of "Freon" 503 azeotrope[a]	—	—
"Freon" 13	Specialty low temperature applications, usually in "cascade" systems	—	—
"Freon" 116	—	—	Dielectric fluid
"Freon" 13B1	Intermediate between "Freon" 13 and "Freon" 22 for medium to low temperature applications. Not extensively used.	—	Efficient fire extinguishant (Halon® 1301) especially suited for automatic protection of materials subject to water damage and of areas occupied by personnel
"Freon" 22	Household and commercial refrigeration and air conditioning applications. Permits use of smaller equipment. Component of azeotropes[a].	—	—
"Freon" 115	Used as an azeotrope component in "Freon" 502[a].	Accepted as a food propellant by the FDA, this material is well suited for food aerosols and finds use in fat emulsion food whips. Good foam stability with absence of odor or taste.	Dielectric fluid, an economic replacement for "Freon" 116 in most dielectric applications.
"Freon" 12	Most widely used refrigerant in household, automotive and commercial refrigeration and air conditioning systems. Also as a component of azeotropes[a] and, in high purity form ("Freon" freezant) approved as a direct contact freezing agent for foods.	Most widely used high pressure propellant for non food use. Blends with "Freon" 11 and "Freon" 114 are widely used.	Blowing agent for foamed plastics applications[d]. Dielectric gas
"Freon" 114	In large industrial process cooling and air conditioning systems using multi stage centrifugal compressors.	Low pressure propellant, alternative to "Freon" 11, having poorer solubility properties and less odor. Especially used in personal products	Blowing agent for foamed plastics.
"Freon" 21	—	—	Heat transfer fluid
"Freon" 11	Widely used in centrifugal compressors for industrial and commercial air conditioning systems and for industrial cooling of process water or brine. Low viscosity and freezing point permit use as a low temperature cooling liquid.	Most widely used low pressure propellant for non-food use. Does not provide adequate pressure alone, so is almost entirely used in blends with "Freon" 12(c).	Occasionally used as a solvent ("Freon" MF). Blowing agent for foamed plastics[d]
"Freon" 113	In commercial and industrial air conditioning and process water or brine chilling using centrifugal compressors, particularly in small tonnage applications.	Solvent in some aerosol formulations, usually propelled with "Freon" 12	Extensively as a solvent ("Freon" TF) alone and in special purpose formulations for a wide range of critical cleaning needs. In cutting fluid formulations, Valclene dry cleaning, etc.

(a) A number of azeotropes ("Freon" 500, "Freon" 502 etc.) are available for refrigeration use. Bulletins describing the composition, properties and uses of these mixtures are available on request.

(b) Normally aerosol propellants are blended to give the required vapor pressure and solubility requirements.

(c) "Freon" 11 S is a stabilized grade frequently used in formulations containing alcohols or water.

(d) "Freon" 12 and "Freon" 11 are widely used as blowing agents for a range of foamed plastics, in which they provide excellent cell structure and the trapped fluorocarbon in closed cell foams significantly improves insulation and recovery properties of the foam.

APPENDIX D

THERMOCOUPLE CALIBRATION

A. EQUIPMENT USED

The equipment used in thermocouple calibration is shown in Figure D.1, and a brief description of each component is given below:

1. Thermocouple Wire

Type-T (copper-constantan) Teflon-coated wire of 0.254 mm (0.01 in.) in diameter was used for all thermocouples.

2. Calibration Bath

A Thermos flask was used as the calibration bath. In order to maintain an isothermal temperature distribution, a motor-driven mixer was used. Observations during the calibration procedure showed bath temperature fluctuations to be ± 0.002 K.

3. Thermocouple Readout

A Hewlett Packard 3497A automatic data acquisition/ control system and a Hewlett Packard 9826 computer were used to read, analyze and record the calibration data.

4. Reference Temperature

A Hewlett Packard 2804A quartz thermometer was used to measure the bath temperature. This quartz thermometer had a resolution of 0.0001 K, while the manufacturer-guaranteed accuracy was better than ± 0.03 K.

B. PREPARATION FOR CALIBRATION

1. Thermocouple Preparation

A total of five thermocouples were prepared for the calibration run. Two thermocouples were made from the beginning of a spool of copper-constantan wire; two were made from the end; and one was made at the mid section using the following procedure:

- The Teflon insulation was removed for a length of about 4 mm from one end of a 1 m long piece of wire and a thermocouple bead was made using a Dynatech Corporation thermocouple welder.
- The other end of the thermocouple wire was connected to the data acquisition and control unit through a junction box.

2. Computer Program

A short computer program (TCAL) was written to accept the thermocouple readings through the data acquisition system and the bath temperature through the digital quartz thermometer. A listing of the computer program, TCAL is provided in Appendix G. This program prints all data and the discrepancy (i.e., the Quartz thermometer reading minus the thermocouple reading) as well as it stores the data on a computer disk. In order to convert the e.m.f. values to temperature, the manufacturer's conversion equation (D.1) was used in the computer program TCAL:

$$T = a_0 + a_1E^1 + a_2E^2 + a_3E^3 + a_4E^4 + a_5E^5 + a_6E^6 + a_7E^7 \tag{D.1}$$

where:

T = temperature (°C)
a_0 = 0.100860910
a_1 = 25727.94369
a_2 = -767345.8295
a_3 = 78025595.81
a_4 = -9247486589
a_5 = 6.97688E+11
a_6 = -2.66192E+13
a_7 = 3.94078E+14
E = thermocouple reading (volts)

C. CALIBRATION PROCEDURE

- Since the temperature measurements during this investigation ranged from 25 °C to 70 °C, thermocouple calibration was performed in this region. First, hot water (at about 75 °C) was added to the thermos flask calibration bath. Following proper mixing, all thermocouple readings were recorded through the data acquisition system and the bath temperature was measured by the quartz thermometer. To obtain lower temperatures, small quantities of cold water were gradually added to the bath.
- The discrepancy (i.e., the Quartz thermometer reading minus the thermocouple reading) was plotted against the thermocouple reading as shown in Figure D.2. The second-order polynomial curve (D.2) shown in this figure was generated using the data from all 5 thermocouples.

$$DCP = -6.7422934E\text{-}02 + 9.0277043E\text{-}03\ T - 9.3259917E\text{-}05\ T^2 \quad (D.2)$$

where:

T = thermocouple reading (from equation D.1) (°C)
DCP = discrepancy (K)

Thus, the corrected temperature values were obtained using equation (D.3):

$$T_b = DCP + T \quad (D.3)$$

where:

T_b = actual calculated temperature (°C)

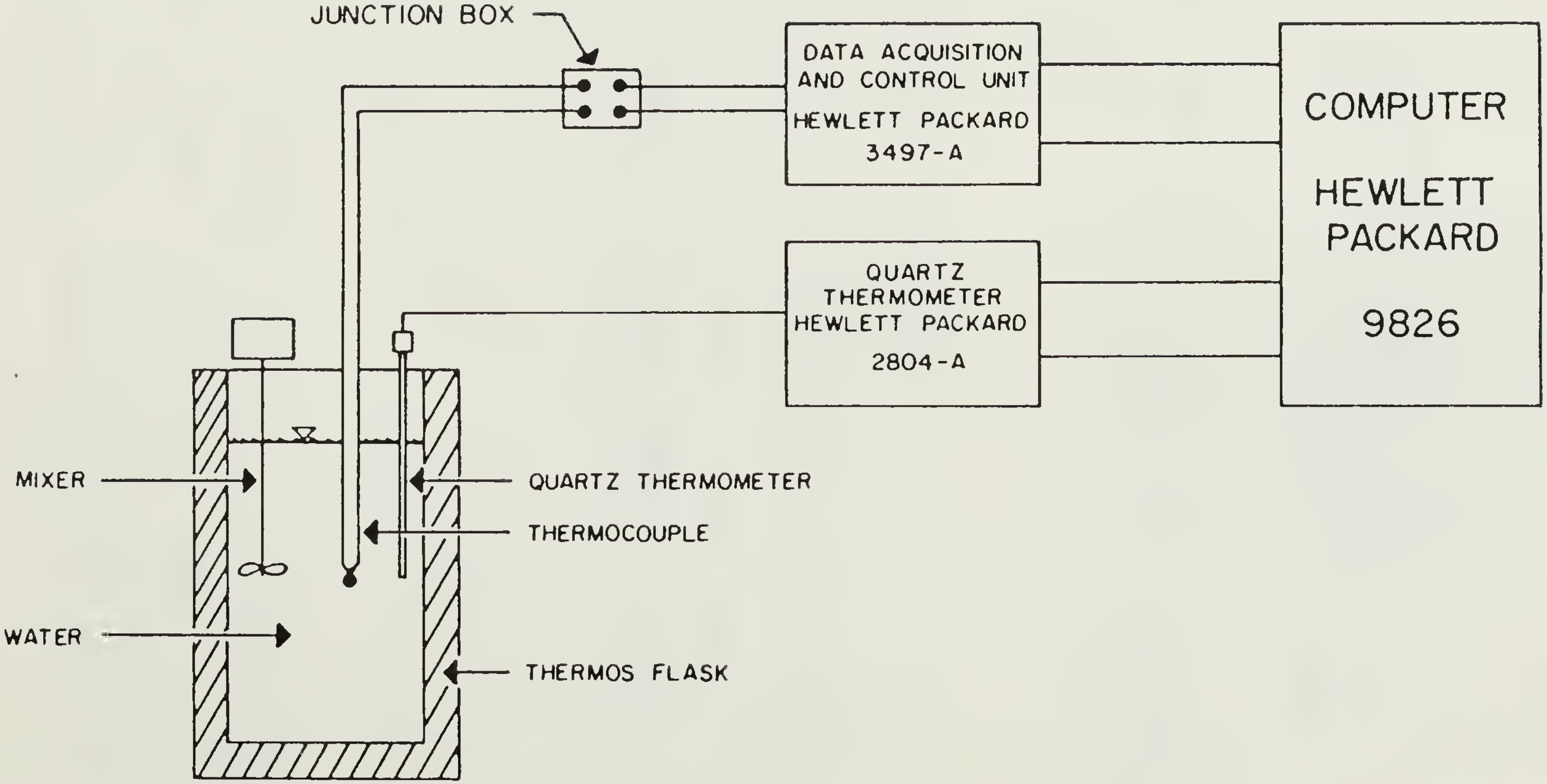
JUNCTION BOX
DATA ACQUISITION AND CONTROL UNIT HEWLETT PACKARD 3497-A
COMPUTER HEWLETT PACKARD 9826
QUARTZ THERMOMETER HEWLETT PACKARD 2804-A
MIXER
QUARTZ THERMOMETER
THERMOCOUPLE
WATER
THERMOS FLASK

Figure D.1 Schematic of the Calibration Devices.

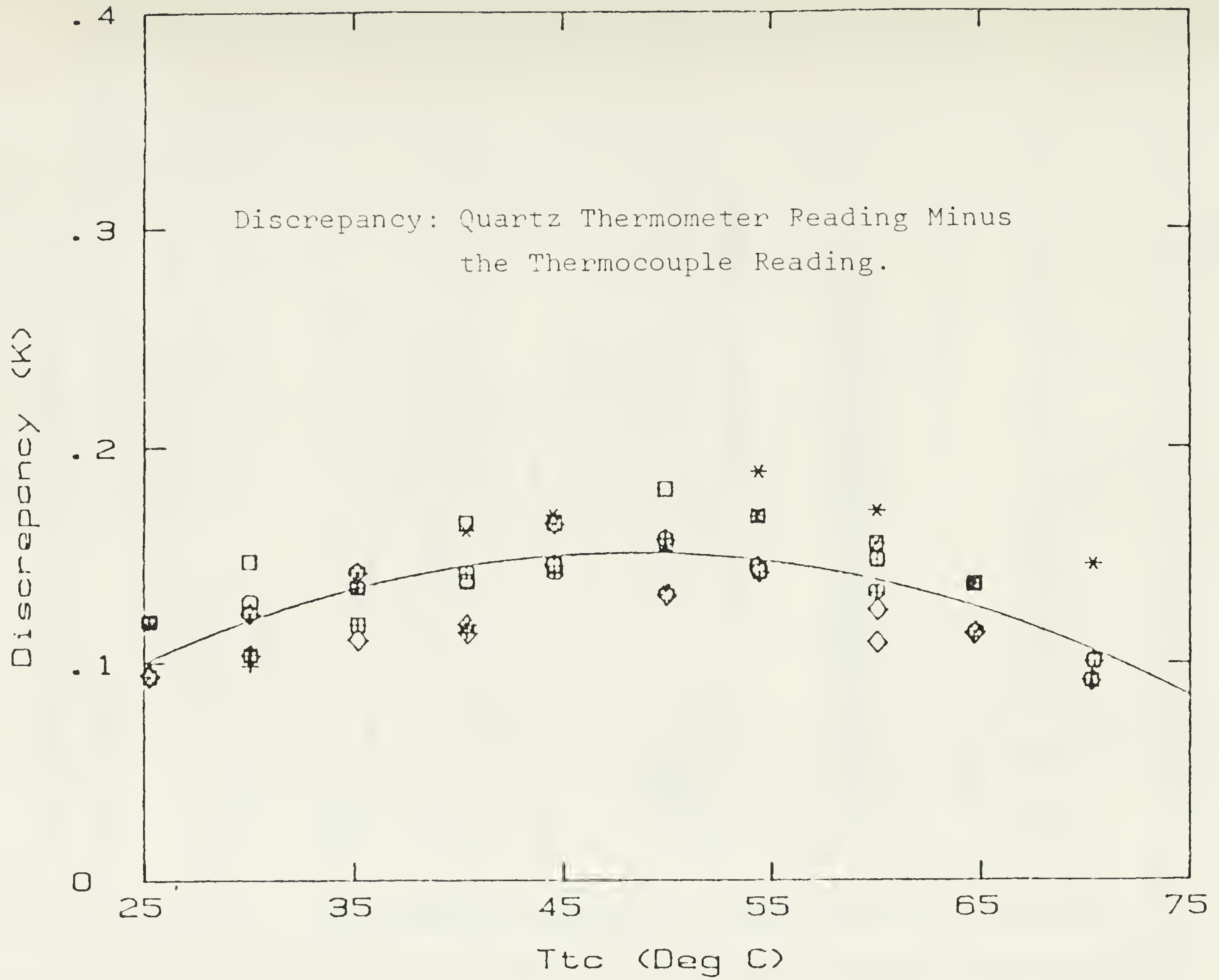

Figure D.2 Thermocouple Calibration Curve.

APPENDIX E

DATA REDUCTION PROGRAM

```
 1000! FILE NAME: DRPR1
1010! DATE:       October 19, 1984
1020! REVISED:    November 1, 1984
1030!
1040  BEEP
1050  PRINTER IS 1
1060  PRINT USING "4X,""Select option:"""
1070  PRINT USING "6X,""0 Taking data or re-processing previous data"""
1080  PRINT USING "6X,""1 Plotting data"""
1090  INPUT Idp
1100  IF Idp=0 THEN CALL Main
1110  IF Idp=1 THEN CALL Plot
1120  END
1130  SUB Main  .
1140  COM /Cc/ C(7)
1150  DIM Emf(12),T(12),D1a(2),D2a(2),Dia(2),Doa(2),La(2),Lua(2),Kcua(2)
1160  DATA 0.10086091,25727.94369,-767345.8295,78025595.81
1170  DATA -9247486589,6.97688E+11,-2.66192E+13,3.94078E+14
1180  READ C(*)
1190  PRINTER IS 701
1200  CLEAR 709
1210  BEEP
1220  INPUT "ENTER MONTH, DATE AND TIME (MM:DD:HH:MM:SS)",Date$
1230  OUTPUT 709;"TD";Date$
1240  OUTPUT 709;"TD"
1250  ENTER 709;Date$
1260  PRINT
1270  PRINT
1280  PRINT
1290  PRINT "            Month, date and time :";Date$
1300  PRINT
1310  PRINT USING "10X,""NOTE: Program name : DRP"""
1320  BEEP
1330  INPUT "ENTER DISK NUMBER",Dn
1340  PRINT USING "16X,""Disk number  = "",ZZ";Dn
1350  BEEP
1360  INPUT "ENTER INPUT MODE (0=3054A,1=FILE)",Im
1370  IF Im=0 THEN
1380  BEEP
1390  INPUT "GIVE A NAME FOR THE RAW DATA FILE",D2_file$
1400  PRINT USING "16X,""New file name: "",14A";D2_file$
1410  CREATE BDAT D2_file$,20
1420  ASSIGN @File2 TO D2_file$
1430!
1440! DUMMY FILE UNTIL Nrun KNOWN
1450  D1_file$="DUMMY"
1460  CREATE BDAT D1_file$,20
1470  ASSIGN @File1 TO D1_file$
1480  OUTPUT @File1;Date$
1490  BEEP
1500  INPUT "GIVE A NAME FOR THE PLOT FILE",P_file$
1510  CREATE BDAT P_file$,5
1520  ASSIGN @Plot TO P_file$
1530  BEEP
1540  INPUT "ENTER NUMBER OF DEFECTIVE TCS (0=DEFAULT)",Idtc
1550  IF Idtc=0 THEN
1560  Ldtc1=0
1570  Ldtc2=0
1580  PRINT USING "16X,""No defective TCs exist"""
1590  END IF
```

```
1600  IF Idtc=1 THEN
1610  BEEP
1620  INPUT "ENTER DEFECTIVE TC LOCATION".Ldtc1
1630  PRINT USING "16X.""TC is defective at location "".D":Ldtc1
1640  Ldtc2=0
1650  END IF
1660  IF Idtc=2 THEN
1670  BEEP
1680  INPUT "ENTER DEFECTIVE TC LOCATIONS".Ldtc1.Ldtc2
1690  PRINT USING "16X.""TC are defective at locations "".D.4X.D":Ldtc1.Ldtc2
1700  END IF
1710  IF Idtc>2 THEN
1720  BEEP
1730  PRINTER IS 1
1740  BEEP
1750  PRINT "INVALID ENTRY"
1760  PRINTER IS 701
1770  GOTO 1530
1780  END IF
1790  OUTPUT @File1:Ldtc1.Ldtc2
1800! Im=1 option
1810  ELSE
1820  BEEP
1830  INPUT "GIVE THE NAME OF THE EXISTING DATA FILE".D2_file$
1840  PRINT USING "16X.""Old file name: "".14A":D2_file$
1850  ASSIGN @File2 TO D2_file$
1860  ENTER @File2:Nrun
1870  ENTER @File2;Dold$
1880  BEEP
1890  INPUT "GIVE A NAME FOR PLOT FILE".P_file$
1900  CREATE BDAT P_file$.5
1910  ASSIGN @Plot TO P_file$
1920  PRINT USING "16X.""This data set taken on : "".14A":Dold$
1930  ENTER @File2:Ldtc1.Ldtc2
1940  PRINT USING "16X.""Thermocouples were defective at locations:"".2(3D.4X)":
Ldtc1.Ldtc2
1950  ENTER @File2:Itt
1960  END IF
1970  PRINTER IS 1
1980  IF Im=0 THEN
1990  BEEP
2000  PRINT USING "4X.""Select tube type"""
2010  PRINT USING "6X.""0=Smooth 4 inch Ref"""
2020  PRINT USING "6X.""1=Smooth 4 inch soft Cu"""
2030  PRINT USING "6X.""2=Smooth 8 inch soft Cu"""
2040  INPUT Itt
2050  IF Itt>2 THEN
2060  BEEP
2070  PRINT "INVALID ENTRY"
2080  GOTO 2000
2090  END IF
2100  OUTPUT @File1:Itt
2110  END IF
2120  PRINTER IS 701
2130  PRINT USING "16X.""Tube Type is : "".D":Itt
2140  BEEP
2150  INPUT "ENTER OUTPUT VERSION (0=LONG.1=SHORT)".Iov
2160!
2170! D1=Diameter at thermocouple positions
2180  DATA .011125,.0111125,.01143
2190  READ Dia(*)
```

```
2200  D1=D1a(Itt)
2210!
2220! D2=Diameter of test section to the base of fins
2230  DATA .015875,.015875,.015875
2240  READ D2a(*)
2250  D2=D2a(Itt)
2260!
2270! Di=Inside diameter of unenhanced ends
2280  DATA .0127,.0127,.0127
2290  READ Dia(*)
2300  Di=Dia(Itt)
2310!
2320! Do=Outside diameter of unenhanced ends
2330  DATA .015875,.015875,.015875
2340  READ Doa(*)
2350  Do=Doa(Itt)
2360!
2370! L=Length of enhanced surface
2380  DATA .1016,.1016,.2032
2390  READ La(*)
2400  L=La(Itt)
2410!
2420! Lu=Length of unenhanced surface at the ends
2430  DATA .0254,.0254,.0762
2440  READ Lua(*)
2450  Lu=Lua(Itt)
2460!
2470! Kcu=Thermal Conductivity of tube
2480  DATA 398,344,344
2490  READ Kcua(*)
2500  Kcu=Kcua(Itt)
2510  A=PI*(Do 2-Di`2)/4
2520  P=PI*Do
2530  J=1
2540  Sx=0
2550  Sy=0
2560  Sxs=0
2570  Sxy=0
2580 Repeat: !
2590  IF Im=0 THEN
2600  ON KEY 0,15 RECOVER 2580
2610  PRINTER IS 1
2620  PRINT USING "4X,""SELECT OPTION"""
2630  PRINT USING "9X,""0=TAKE DATA"""
2640  PRINT USING "9X,""1=SET HEAT FLUX"""
2650  PRINT USING "9X,""2=SET Tsat"""
2660  PRINT USING "4X,""NOTE: KEY 0 = ESCAPE"""
2670  BEEP
2680  INPUT Ido
2690  IF Ido=0 THEN 3680
2700!
2710! LOOP TO SET HEAT FLUX
2720  IF Ido=1 THEN
2730  OUTPUT 709;"AR AF62 AL63 VR5"
2740  BEEP
2750  INPUT "ENTER DESIRED Qdp",Dqdp
2760  PRINT USING "4X,""DESIRED Qdp  ACTUAL Qdp"""
2770  Err=1000
2780  FOR I=1 TO 2
2790  OUTPUT 709;"AS SA"
2800  Sum=0
```

```
2810  FOR Ji=1 TO 5
2820  ENTER 709:E
2830  Sum=Sum+E
2840  NEXT Ji
2850  IF I=1 THEN Volt=Sum*5
2860  IF I=2 THEN Amp=E
2870  NEXT I
2880  Aqdp=Volt*Amp/(PI*D2*L)
2890  IF ABS(Aqdp-Dqdp)>Err THEN
2900  IF Aqdp>Dqdp THEN
2910  BEEP 4000,.2
2920  BEEP 4000,.2
2930  BEEP 4000,.2
2940  ELSE
2950  BEEP 250,.2
2960  BEEP 250,.2
2970  BEEP 250,.2
2980  END IF
2990  PRINT USING "4X,MZ.3DE,2X,MZ.3DE":Dqdp,Aqdp
3000  WAIT 2
3010  GOTO 2780
3020  ELSE
3030  BEEP
3040  PRINT USING "4X,MZ.3DE,2X,MZ.3DE":Dqdp,Aqdp
3050  Err=500
3060  WAIT 2
3070  GOTO 2780
3080  END IF
3090  END IF
3100!
3110! LOOP TO SET Tsat
3120  IF Ido=2 THEN
3130  OUTPUT 709:"AR AF33 AL33 VR5"
3140  OUTPUT 709:"AS SA"
3150  BEEP
3160  INPUT "ENTER DESIRED Tsat",Dtld
3170  PRINT USING "4X,""D Tsat   A Tsat   Diff      Psat"""
3180  Old=0
3190  ENTER 709:Eliq
3200  Atld=FNTvsv(Eliq)
3210  Psat=FNPsat(Atld)
3220  IF ABS(Atld-Dtld)>.2 THEN
3230  IF Atld>Dtld THEN
3240  BEEP 4000,.2
3250  BEEP 4000,.2
3260  BEEP 4000,.2
3270  ELSE
3280  BEEP 250,.2
3290  BEEP 250,.2
3300  BEEP 250,.2
3310  END IF
3320  Err=Atld-Old
3330  Old=Atld
3340  PRINT USING "4X,4(MDD.DD,3X)":Dtld,Atld,Err,Psat
3350  WAIT 2
3360  GOTO 3190
3370  ELSE
3380  IF ABS(Atld-Dtld)>.1 THEN
3390  IF Atld>Dtld THEN
3400  BEEP 3000,.2
3410  BEEP 3000,.2
```

```
3420  ELSE
3430  BEEP 800,.2
3440  BEEP 800,.2
3450  END IF
3460  Err=Atld-Old
3470  Old=Atld
3480  PRINT USING "4X,4(MDD.DD,3X)";Dtld,Atld,Err,Psat
3490  WAIT 2
3500  GOTO 3190
3510  ELSE
3520  BEEP
3530  Err=Atld-Old
3540  Old=Atld
3550  PRINT USING "4X,4(MDD.DD,3X)";Dtld,Atld,Err,Psat
3560  WAIT 2
3570  GOTO 3190
3580  END IF
3590  END IF
3600  END IF
3610! ERROR TRAP FOR Ido OUT OF BOUNDS
3620  IF Ido>2 THEN
3630  BEEP
3640  GOTO 2620
3650  END IF
3660!
3670! TAKE DATA IF Im=0 LOOP
3680  BEEP
3690  INPUT "ENTER BULK OIL %",Bop
3700  OUTPUT 709;"AR AF25 AL36 VR5"
3710  FOR I=1 TO 12
3720  OUTPUT 709;"AS SA"
3730  Sum=0
3740  FOR Ji=1 TO 20
3750  ENTER 709;E
3760  Sum=Sum+E
3770  NEXT Ji
3780  Emf(I)=Sum/20
3790  NEXT I
3800  OUTPUT 709;"AR AF62 AL63 VR5"
3810  FOR I=1 TO 2
3820  OUTPUT 709;"AS SA"
3830  Sum=0
3840  FOR Ji=1 TO 20
3850  ENTER 709;E
3860  Sum=Sum+E
3870  NEXT Ji
3880  IF I=1 THEN Vr=Sum/20
3890  IF I=2 THEN Ir=Sum/20
3900  NEXT I
3910  ELSE
3920  ENTER @File2;Bop,Told$,Emf(*),Vr,Ir
3930  END IF
3940!
3950! CONVERT emf'S TO TEMP,VOLT,CURRENT
3960  Twa=0
3970  FOR I=1 TO 12
3980  IF Idtc>0 THEN
3990  IF I=Ldtc1 OR I=Ldtc2 THEN
4000  T(I)=-99.99
4010  GOTO 4060
4020  END IF
```

```
4030  END IF
4040  T(I)=FNTvsv(Emf(I))
4050  IF I<9 THEN Twa=Twa+T(I)
4060  NEXT I
4070  Twa=Twa/(8-Idtc)
4080  Tld=T(9)
4090  Tv=(T(10)+T(11))/2
4100  Tsump=T(12)
4110  Amp=Ir
4120  Volt=Vr*25
4130  Q=Volt*Amp
4140  IF Itt=0 THEN
4150  Kcu=FNKcu(Twa)
4160  ELSE
4170  Kcu=Kcua(Itt)
4180  END IF
4190!
4200! FOURIER CONDUCTION EQUATION WITH CONTACT RESISTANCE NEGLECTED
4210  Tw=Twa-Q*LOG(D2/D1)/(2*PI*Kcu*L)
4220  Thetab=Tw-Tld
4230  IF Thetab<0 THEN
4240  BEEP
4250  INPUT "TWALL<TSAT (0=CONTINUE, 1=END)",Iev
4260  IF Iev=0 THEN GOTO 2590
4270  IF Iev=1 THEN 5030
4280  END IF
4290!
4300! COMPUTE VARIOUS PROPERTIES
4310  Tfilm=FNTfilm(Tw,Tld)
4320  Rho=FNRho(Tfilm)
4330  Mu=FNMu(Tfilm)
4340  K=FNK(Tfilm)
4350  Cp=FNCp(Tfilm)
4360  Beta=FNBeta(Tfilm)
4370  Ni=Mu/Rho
4380  Alpha=K/(Rho*Cp)
4390  Pr=Ni/Alpha
4400  Psat=FNPsat(Tld)
4410!
4420! COMPUTE HEAT TRANSFER COEFFICIENT
4430  Hbar=190
4440  Fe=(Hbar*P/(Kcu*A))`.5*Lu
4450  Tanh=FNTanh(Fe)
4460  Theta=Thetab*Tanh/Fe
4470! PRINTER IS 701
4480! PRINT USING "4X,7(1X,MZ.3DE)";Hbar,Fe,Tanh,Thetab,Theta,Beta,Ni
4490  Xx=(9.81*Beta*Thetab*Do`3*Tanh/(Fe*Ni*Alpha)) .166667
4500  Yy=(1+(.559/Pr)`(9/16)) (8/27)
4510  Hbarc=K/Do*(.6+.387*Xx/Yy)`2
4520  IF ABS((Hbar-Hbarc)/Hbarc)>.001 THEN
4530  Hbar=(Hbar+Hbarc)*.5
4540  GOTO 4440
4550  END IF
4560  Ql=(Hbar*P*Kcu*A) .5*Thetab*Tanh
4570  Qc=Q-2*Ql
4580  As=PI*D2*L
4590  Qdp=Qc/As
4600  Htube=Qdp/Thetab
4610!
4620! RECORD TIME OF DATA TAKING
4630  IF Im=0 THEN
```

```
4640  OUTPUT 709;"TD"
4650  ENTER 709;Told$
4660  END IF
4670!
4680! OUTPUT DATA TO PRINTER
4690  PRINTER IS 701
4700  PRINT
4710  IF Iov=0 THEN
4720  PRINT USING "11X,""Data Set Number = "",DDD,2X,""Bulk Oil % = "",DD.D,5X,1
4A":J,Bop,Told$
4730  PRINT
4740  PRINT USING "11X,""TC No:    1       2       3       4       5       6       7
   8"""""
4750  PRINT USING "11X,""Temp :"",8(1X,MDD.DD)":T(1),T(2),T(3),T(4),T(5),T(6),T(
7),T(8)
4760  PRINT USING "11X,"" Twa     Tliqd    Thetab      Tvapr  Psat    Tsump"""""
4770  PRINT USING "11X,2(MDD.DD,1X),MZ.DDDE,1X,2(1X,MDD.DD),2X,MDD.D":Twa,Tld,Th
etab,Tv,Psat,Tsump
4780  PRINT USING "11X,"" Htube       Qdp"""""
4790  PRINT USING "11X,MZ.3DE,1X,MZ.3DE":Htube,Qdp
4800  END IF
4810  IF Iov=1 THEN
4820  IF J=1 THEN
4830  PRINT USING "11X,""RUN No Oil%  Tsat   Htube      Qdp         Thetab"""""
4840  END IF
4850  PRINT USING "12X,DDD,4X,DD,2X,MDD.DD,3(1X,MZ.DDDE)":J,Bop,Tld,Htube,Qdp,Th
etab
4860  END IF
4870  IF Im=0 THEN
4880  BEEP
4890  INPUT "OK TO STORE THIS DATA SET (1=Y,0=N)?",Ok
4900  END IF
4910  IF Ok=1 OR Im=1 THEN J=J+1
4920  IF Ok=1 AND Im=0 THEN OUTPUT @File1:Bop,Told$,Emf(*),Vr,Ir
4930  IF Im=1 OR Ok=1 THEN OUTPUT @Plot:Qdp,Thetab
4940  IF Im=0 THEN
4950  BEEP
4960  INPUT "WILL THERE BE ANOTHER RUN (1=Y,0=N)?",Go_on
4970  Nrun=J
4980  IF Go_on<>1 THEN 5030
4990  IF Go_on=1 THEN Repeat
5000  ELSE
5010  IF J<Nrun+1 THEN Repeat
5020  END IF
5030  IF Im=0 THEN
5040  BEEP
5050  PRINT
5060  PRINT USING "10X,""NOTE: "",ZZ,"" data runs were stored in file "",10A":J-
1,D2_file$
5070  ASSIGN @File1 TO *
5080  OUTPUT @File2:Nrun-1
5090  ASSIGN @File1 TO D1_file$
5100  ENTER @File1:Date$,Ldtc1,Ldtc2,Itt
5110  OUTPUT @File2:Date$,Ldtc1,Ldtc2,Itt
5120  FOR I=1 TO Nrun-1
5130  ENTER @File1:Bop,Told$,Emf(*),Vr,Ir
5140  OUTPUT @File2:Bop,Told$,Emf(*),Vr,Ir
5150  NEXT I
5160  ASSIGN @File1 TO *
5170  PURGE "DUMMY"
5180  END IF
```

```
5190  BEEP
5200  PRINT
5210  PRINT USING "10X,""NOTE: "",ZZ,"" X-Y pairs were stored in plot data file
"",10A":J-1,P_file$
5220  ASSIGN @File2 TO *
5230  ASSIGN @Plot TO *
5240  BEEP
5250  INPUT "LIKE TO PLOT DATA (1=Y,0=N)?",Ok
5260  IF Ok=1 THEN
5270  CALL Plot
5280  END IF
5290  SUBEND
5300!
5310! CURVE FITS OF PROPERTY FUNCTIONS
5320  DEF FNKcu(T)
5330! OFHC COPPER 250 TO 300 K
5340  Tk=T+273.15       !C TO K
5350  K=434-.112*Tk
5360  RETURN K
5370  FNEND
5380  DEF FNMu(T)
5390! 170 TO 360 K CURVE FIT OF VISCOUSITY
5400  Tk=T+273.15       !C TO K
5410  Mu=EXP(-4.4636+(1011.47/Tk))*1.0E-3
5420  RETURN Mu
5430  FNEND
5440  DEF FNCp(T)
5450! 180 TO 400 K CURVE FIT OF Cp
5460  Tk=T+273.15       !C TO K
5470  Cp=.40188+1.65007E-3*Tk+1.51494E-6*Tk 2-6.67853E-10*Tk`3
5480  RETURN Cp
5490  FNEND
5500  DEF FNRho(T)
5510  Tk=T+273.15       !C TO K
5520  X=1-(1.8*Tk/753.95)  !K TO R
5530  Ro=36.32+61.146414*X`(1/3)+16.418015*X+17.476838*X .5+1.119828*X`2
5540  RETURN Ro
5550  FNEND
5560  DEF FNPr(T)
5570  Pr=FNCp(T)*FNMu(T)/FNK(T)
5580  RETURN Pr
5590  FNEND
5600  DEF FNK(T)
5610! T<360 K WITH T IN C
5620  K=.071-.000261*T
5630  RETURN K
5640  FNEND
5650  DEF FNTanh(X)
5660  P=EXP(X)
5670  Q=EXP(-X)
5680  Tanh=(P-Q)/(P+Q)
5690  RETURN Tanh
5700  FNEND
5710  DEF FNTvsv(V)
5720  COM /Cc/ C(7)
5730  T=C(0)
5740  FOR I=1 TO 7
5750  T=T+C(I)*V`I
5760  NEXT I
5770  T=T-6.7422934E-2+T*(9.0277043E-3-T*(-9.3259917E-5))
```

```
5780  RETURN T
5790  FNEND
5800  DEF FNBeta(T)
5810  Rop=FNRho(T+.1)
5820  Rom=FNRho(T-.1)
5830  Beta=-2/(Rop+Rom)*(Rop-Rom)/.2
5840  RETURN Beta
5850  FNEND
5860  DEF FNTfilm(Tw,Tld)
5870  Tfilm=(Tw+Tld)/2
5880  RETURN Tfilm
5890  FNEND
5900  DEF FNPsat(Tc)
5910! 0 TO 80 deg F CURVE FIT OF Psat
5920  Tf=1.8*Tc+32
5930  Pa=5.945525+Tf*(.15352082+Tf*(1.4840963E-3+Tf*9.6150671E-6))
5940  Pg=Pa-14.7
5950  IF Pg>0 THEN        ! +=PSIG,-=in Hg
5960  Psat=Pg
5970  ELSE
5980  Psat=Pg*29.92/14.7
5990  END IF
6000  RETURN Psat
6010  FNEND
6020  SUB Plot
6030  DIM C(9)
6040  INTEGER Ii
6050  PRINTER IS 1
6060  Idv=1
6070  BEEP
6080  INPUT "LIKE DEFAULT VALUES FOR PLOT (1=Y,0=N)?",Idv
6090  BEEP
6100  PRINT USING "4X,""Select Option:"""
6110  PRINT USING "4X,""0  q versus delta-T"""
6120  PRINT USING "4X,""1  h versus delta-T"""
6130  PRINT USING "4X,""2  h versus q"""
6140  PRINT USING "4X,""3  h-ratio versus delta-T"""
6150  INPUT Opo
6160  IF Opo=3 THEN
6170  BEEP
6180  INPUT "SELECT TUBE DIAMETER (0=.75,1=1.0 IN)",Itd
6190  END IF
6200  PRINTER IS 705
6210  IF Idv<>1 THEN
6220  BEEP
6230  INPUT "ENTER NUMBER OF CYCLES FOR X-AXIS",Cx
6240  BEEP
6250  INPUT "ENTER NUMBER OF CYCLES FOR Y-AXIS",Cy
6260  BEEP
6270  INPUT "ENTER MIN X-VALUE (MULTIPLE OF 10)",Xmin
6280  BEEP
6290  INPUT "ENTER MIN Y-VALUE (MULTIPLE OF 10)",Ymin
6300  ELSE
6310  Cy=3
6320  IF Opo=0 THEN
6330  Cx=2
6340  Xmin=1
6350  Ymin=1000
6360  END IF
6370  IF Opo=1 THEN
6380  Cx=2
```

```
6390  Xmin=1
6400  Ymin=100
6410  END IF
6420  IF Opo=2 THEN
6430  Cx=3
6440  Xmin=1000
6450  Ymin=100
6460  END IF
6470  IF Opo=3 THEN
6480  Cx=2
6490  Cy=3
6500  Xmin=1
6510  Ymin=1
6520  END IF
6530  END IF
6540  BEEP
6550  PRINT "IN:SP1:IP 2300.1800.8300.6800:"
6560  PRINT "SC 0,100,0.100:TL 2,0:"
6570  Sfx=100/Cx
6580  Sfy=100/Cy
6590  PRINT "PU 0,0 PD"
6600  Nn=9
6610  FOR I=1 TO Cx+1
6620  Xat=Xmin*10^(I-1)
6630  IF I=Cx+1 THEN Nn=1
6640  FOR J=1 TO Nn
6650  IF J=1 THEN PRINT "TL 2 0"
6660  IF J=2 THEN PRINT "TL 1 0"
6670  Xa=Xat*J
6680  X=LGT(Xa/Xmin)*Sfx
6690  PRINT "PA":X.",0: XT:"
6700  NEXT J
6710  NEXT I
6720  PRINT "PA 100,0:PU:"
6730  PRINT "PU PA 0,0 PD"
6740  Nn=9
6750  FOR I=1 TO Cy+1
6760  Yat=Ymin*10^(I-1)
6770  IF I=Cy+1 THEN Nn=1
6780  FOR J=1 TO Nn
6790  IF J=1 THEN PRINT "TL 2 0"
6800  IF J=2 THEN PRINT "TL 1 0"
6810  Ya=Yat*J
6820  Y=LGT(Ya/Ymin)*Sfy
6830  PRINT "PA 0,":Y,"YT"
6840  NEXT J
6850  NEXT I
6860  PRINT "PA 0,100 TL 0 2"
6870  Nn=9
6880  FOR I=1 TO Cx+1
6890  Xat=Xmin*10 (I-1)
6900  IF I=Cx+1 THEN Nn=1
6910  FOR J=1 TO Nn
6920  IF J=1 THEN PRINT "TL 0 2"
6930  IF J>1 THEN PRINT "TL 0 1"
6940  Xa=Xat*J
6950  X=LGT(Xa/Xmin)*Sfx
6960  PRINT "PA":X.",100: XT"
6970  NEXT J
6980  NEXT I
```

```
6990  PRINT "PA 100.100 PU PA 100.0 PD"
7000  Nn=9
7010  FOR I=1 TO Cy+1
7020  Yat=Ymin*10^(I-1)
7030  IF I=Cy+1 THEN Nn=1
7040  FOR J=1 TO Nn
7050  IF J=1 THEN PRINT "TL 0 2"
7060  IF J>1 THEN PRINT "TL 0 1"
7070  Ya=Yat*J
7080  Y=LGT(Ya/Ymin)*Sfy
7090  PRINT "PD PA 100,",Y,"YT"
7100  NEXT J
7110  NEXT I
7120  PRINT "PA 100.100 PU"
7130  PRINT "PA 0.-2 SR 1.5.2"
7140  Ii=LGT(Xmin)
7150  FOR I=1 TO Cx+1
7160  Xa=Xmin*10^(I-1)
7170  X=LGT(Xa/Xmin)*Sfx
7180  PRINT "PA";X,",0;"
7190  IF Ii>=0 THEN PRINT "CP -2.-2;LB10;PR -2.2;LB";Ii;""
7200  IF Ii<0 THEN PRINT "CP -2.-2;LB10;PR 0.2;LB";Ii;""
7210  Ii=Ii+1
7220  NEXT I
7230  PRINT "PU PA 0.0"
7240  Ii=LGT(Ymin)
7250  Y10=10
7260  FOR I=1 TO Cy+1
7270  Ya=Ymin*10^(I-1)
7280  Y=LGT(Ya/Ymin)*Sfy
7290  PRINT "PA 0,":Y,""
7300  PRINT "CP -4.-.25;LB10;PR -2.2;LB":Ii:""
7310  Ii=Ii+1
7320  NEXT I
7330  IF Idv<>1 THEN
7340  BEEP
7350  INPUT "ENTER X-LABEL".Xlabel$
7360  BEEP
7370  INPUT "ENTER Y-LABEL".Ylabel$
7380  ELSE
7390  IF Opo=0 THEN
7400  Xlabel$="Tw-Tsat (K)"
7410  Ylabel$="q (W/m^2)"
7420  END IF
7430  IF Opo=1 THEN
7440  Xlabel$="Tw-Tsat (K)"
7450  Ylabel$="h (W/m^2.K)"
7460  END IF
7470  IF Opo=2 THEN
7480  Xlabel$="q (W/m^2)"
7490  Ylabel$="h (W/m^2.K)"
7500  END IF
7510  IF Opo=3 THEN
7520  Xlabel$="Tw-Tsat (K)"
7530  Ylabel$="h(enh)/h(smooth)"
7540  END IF
7550  END IF
7560  PRINT "SR 1.5.2;PU PA 50.-16 CP":-LEN(Xlabel$)/2:"0:LB":Xlabel$:""
7570  PRINT "PA -14.50 CP 0.":-LEN(Ylabel$)/2*5/6;"DI 0.1:LB";Ylabel$:""
7580  PRINT "CP 0.0 DI"
7590 Repeat:!
```

```
7600  BEEP
7610  INPUT "WANT TO PLOT DATA FROM A FILE (1=Y.0=N)?".Ok
7620  IF Ok=1 THEN
7630  BEEP
7640  INPUT "ENTER THE NAME OF THE DATA FILE".D_file$
7650  ASSIGN @File TO D_file$
7660  BEEP
7670  BEEP
7680  INPUT "ENTER THE BEGINNING RUN NUMBER".Md
7690  BEEP
7700  INPUT "ENTER THE NUMBER OF X-Y PAIRS STORED".Npairs
7710  BEEP
7720  PRINTER IS 1
7730  PRINT USING "4X.""Select a symbol:"""
7740  PRINT USING "4X.""1  Star      2  Plus sign"""
7750  PRINT USING "4X.""3  Circle    4  Square"""
7760  PRINT USING "4X.""5  Rombus"""
7770  PRINT USING "4X.""6  Right-side-up triangle"""
7780  PRINT USING "4X.""7  Up-side-down triangle"""
7790  INPUT Sym
7800  PRINTER IS 705
7810  PRINT "PU DI"
7820  IF Sym=1 THEN PRINT "SM*"
7830  IF Sym=2 THEN PRINT "SM+"
7840  IF Sym=3 THEN PRINT "SMo"
7850  IF Md>1 THEN
7860  FOR I=1 TO (Md-1)
7870  ENTER @File;Ya.Xa
7880  NEXT I
7890  END IF
7900  FOR I=1 TO Npairs
7910  ENTER @File;Ya.Xa
7920  IF Opo=1 THEN Ya=Ya/Xa
7930  IF Opo=2 THEN
7940  Q=Ya
7950  Ya=Ya/Xa
7960  Xa=Q
7970  END IF
7980  IF Opo=3 THEN Ya=Ya/FNHsmooth(Xa.Itd)
7990  X=LGT(Xa/Xmin)*Sfx
8000  Y=LGT(Ya/Ymin)*Sfy
8010  IF Sym>3 THEN PRINT "SM"
8020  IF Sym<4 THEN PRINT "SR 1.4.2.4"
8030  PRINT "PA".X.Y.""
8040  IF Sym>3 THEN PRINT "SR 1.2.1.6"
8050  IF Sym=4 THEN PRINT "UC2.4.99.0.-8.-4.0.0.8.4.0:"
8060  IF Sym=5 THEN PRINT "UC3.0.99.-3.-6.-3.6.3.6.3.-6:"
8070  IF Sym=6 THEN PRINT "UC0.5.3.99.3.-8.-6.0.3.8:"
8080  IF Sym=7 THEN PRINT "UC0.-5.3.99.-3.8.6.0.-3.-8:"
8090  NEXT I
8100  BEEP
8110  ASSIGN @File TO *
8120  GOTO 7600
8130  END IF
8140  PRINT "PU SM"
8150  BEEP
8160  INPUT "WANT TO PLOT A POLYNOMIAL (1=Y.0=N)?".Go_on
8170  IF Go_on=1 THEN
8180  BEEP
8190  INPUT "ENTER LOWER AND UPPER X-LIMITS".Xll.Xlu
```

```
8200  FOR Xx=0 TO Cx STEP Cx/200
8210  Xa=Xmin*10^Xx
8220  IF Xa<Xll OR Xa>Xlu THEN 8290
8230  Ya=FNPoly(Xa)
8240  Y=LGT(Ya/Ymin)*Sfy
8250  X=LGT(Xa/Xmin)*Sfx
8260  IF Y<0 THEN Y=0
8270  IF Y>100 THEN GOTO 8290
8280  PRINT "PA",X,Y,"PD"
8290  NEXT Xx
8300  END IF
8310  PRINT "PU PA 0,0 SP0"
8320  SUBEND
8330  DEF FNHsmooth(X,Itd)
8340  Hs=FNPoly(X)/X
8350  IF Itd=1 THEN Hs=Hs*.83347
8360  RETURN Hs
8370  FNEND
8380  DEF FNPoly(X)
8390  Poly=-4.4123718E+2-X*(6.8123917E+2-X*3.7416863E+2)
8400  RETURN Poly
8410  FNEND
8420  DEF FNPvst(Tsteam)
8430  DIM K(8)
8440  DATA -7.691234564,-26.08023696,-168.1706546,64.23285504,-118.9646225
8450  DATA 4.16711732,20.9750676,1.E9,6
8460  READ K(*)
8470  T=(Tsteam+273.15)/647.3
8480  Sum=0
8490  FOR N=0 TO 4
8500  Sum=Sum+K(N)*(1-T)^(N+1)
8510  NEXT N
8520  Br=Sum/(T*(1+K(5)*(1-T)+K(6)*(1-T)^2))-(1-T)/(K(7)*(1-T)^2+K(8))
8530  Pr=EXP(Br)
8540  P=22120000*Pr
8550  RETURN P
8560  FNEND
```

APPENDIX F

AN EXAMPLE OF REPRESENTATIVE DATA RUN

```
Month, date and time :12:03:16:50:24

NOTE: Program name : DRP
      Disk number  = 02
      Old file name: NH05
      This data set taken on : 11:02:10:36:28
      Thermocouples were defective at locations:  0      0
      Tube Type is : 2

 Data Set Number =   1  Bulk Oil % =  0.0      11:02:11:01:46

 TC No:     1       2       3       4       5       6       7       8
 Temp :  70.05   55.07   61.04   70.23   78.68   60.22   76.21   59.24
  Twa     Tliqd   Thetab      Tvapr  Psat   Tsump
  56.34  10.03   5.560E+01   10.01    3.86  -13.0
  Htube       Qdp
  1.545E+03  8.588E+04

 Data Set Number =   2  Bulk Oil % =  0.0      11:02:11:30:52

 TC No:     1       2       3       4       5       6       7       8
 Temp :  52.64   42.70   45.49   49.95   58.26   44.85   57.78   47.04
  Twa     Tliqd   Thetab      Tvapr  Psat   Tsump
  49.84  10.07   3.939E+01   10.30    3.88  -12.0
  Htube       Qdp
  1.152E+03  4.538E+04

 Data Set Number =   3  Bulk Oil % =  0.0      11:02:11:38:07

 TC No:     1       2       3       4       5       6       7       8
 Temp :  34.63   30.25   31.40   33.49   37.59   31.38   38.00   32.96
  Twa     Tliqd   Thetab      Tvapr  Psat   Tsump
  33.72  10.00   2.357E+01   10.69    3.83  -11.9
  Htube       Qdp
  7.420E+02  1.749E+04

 Data Set Number =   4  Bulk Oil % =  0.0      11:02:11:49:56

 TC No:     1       2       3       4       5       6       7       8
 Temp :  27.30   24.92   25.40   26.60   29.04   25.25   29.30   26.40
  Twa     Tliqd   Thetab      Tvapr  Psat   Tsump
  25.78  10.07   1.663E+01   11.17    3.88  -11.9
  Htube       Qdp
  5.211E+02  8.664E+03

 Data Set Number =   5  Bulk Oil % =  0.0 .    11:02:11:57:02

 TC No:     1       2       3       4       5       6       7       8
 Temp :  22.10   21.03   21.06   21.41   23.11   21.55   23.11   21.65
  Twa     Tliqd   Thetab      Tvapr  Psat   Tsump
  21.88   9.95   1.189E+01   11.47    3.80  -12.1
  Htube       Qdp
  3.094E+02  3.680E+03

 Data Set Number =   6  Bulk Oil % =  0.0      11:02:12:05:46
 TC No:     1       2       3       4       5       6       7       8
 Temp :  18.14   17.58   17.82   17.57   19.16   18.03   18.43   16.84
  Twa     Tliqd    Thetab       Tvapr   Psat    Tsump
  17.95   10.19   7.737E+00     12.00    3.96   -12.2
  Htube       Qdp
  1.728E+02  1.337E+03

 Data Set Number =    7  Bulk Oil % =  0.0      11:02:12:14:18

 TC No:     1       2       3       4       5       6       7       8
 Temp :  15.58   15.08   15.14   14.87   15.66   15.04   15.46   14.55
  Twa     Tliqd    Thetab       Tvapr   Psat    Tsump
  15.17   10.12   5.042E+00     12.34    3.92   -12.2
  Htube       Qdp
  1.221E+02  6.156E+02

NOTE: 07 X-Y pairs were stored in plot data file P05
```

APPENDIX G

LISTING OF CALIBRATION COMPUTER PROGRAM (TCAL)

```
100 ! FILE NAME: TCAL
110 ! REVISED:   October 4, 1984
120 !
130   COM /Cc/ C(7)
140   DIM Emf(4),T(4),D(4)
150   DATA 0,10086091,25727.94369,-767345.8295,78025595.81
160   DATA -9247486589,6.97688E11,-2.66192E13,3.94078E14
170   READ C(*)
180   CLEAR 709
190   BEEP
200   INPUT "ENTER MONTH, DATE AND TIME (MM:DD:HH:MM:SS)",B$
210   J=0
220   OUTPUT 709;"TD";B$
230   OUTPUT 709;"TD"
240   ENTER 709;A$
241   BEEP
242   INPUT "WANT A HARD COPY (1=Y,0=N)?",Ihp
244   IF Ihp=1 THEN PRINTER IS 701
245   IF Ihp=0 THEN PRINTER IS 1
250   PRINT USING "10X,""Month, date and time  = "",14A";A$
260   BEEP
270   INPUT "ENTER INPUT MODE (1=3054A, 2=FILE)",Im
280   IF Im=1 THEN
290   BEEP
300   INPUT "GIVE A NAME FOR DATA FILE",D_file$
310   CREATE BDAT D_file$,20
320   ELSE
330   BEEP
340   INPUT "GIVE NAME OF EXISTING FILE",D_file$
350   BEEP
360   INPUT "ENTER NUMBER OF DATA RUNS STORED",Nrun
361   BEEP
362   INPUT "GIVE A NAME FOR OUTPUT FILE",Ofile$
363   CREATE BDAT Ofile$,10
364   ASSIGN @Fileo TO Ofile$
365   K=1
370   END IF
380   ASSIGN @File TO D_file$
390   IF Im=1 THEN
400   BEEP
410 ! INPUT "ENTER BATH TEMPERATURE",T_bath
411   OUTPUT 713;"T2R2E"
412   WAIT 2
413   ENTER 713;T_bath
420   OUTPUT 709;"AR AF29 AL33 VR1"
430   FOR I=0 TO 4
440   OUTPUT 709;"AS SA"
450   ENTER 709;Emf(I)
460   NEXT I
470   ELSE
480   ENTER @File;T_bath,Emf(*)
490   END IF
500   J=J+1
510   FOR I=0 TO 4
520   T(I)=FNTvsv(ABS(Emf(I)))
530   D(I)=T_bath-T(I)
540   NEXT I
541   IF K=1 THEN
542   FOR L=0 TO 4
544   OUTPUT @Fileo;T(L),D(L)
```

```
545    NEXT L
548    GOTO 700
549    END IF
550    PRINT
560    PRINT USING "10X,""Bath temperature        = "",3D.3D,"" (Deg C)""";T_bath
570    PRINT USING "10X,""Thermocouple readings (Deg C):"""
580    PRINT USING "10X,5(3D.DD,3X),25X";T(*)
590    PRINT USING "10X,""Discrepancies (Deg C):"""
600    PRINT USING "11X,5(MZ.DD,4X),24X";D(*)
610    IF Im=2 THEN 700
620    BEEP
630    INPUT "OK TO ACCEPT THIS SET (1=Y,0=N)?",Oks
640    IF Oks=0 THEN
650    J=J-1
660    GOTO 390
670    ELSE
680    OUTPUT @File;T_bath,Emf(*)
690    END IF
700    BEEP
710    IF Im=1 THEN
720    INPUT "WILL THERE BE ANOTHER SET (1=Y,0=N)?",Go_on
730    IF Go_on=1 THEN 390
740    ELSE
750    IF J<Nrun THEN 390
760    END IF
770    PRINT
780    IF Im=1 THEN
790    PRINT USING "10X,""NOTE: "",DD,"" data sets are stored in file "",14A";J,D
_file$
800    ELSE
810    PRINT USING "10X,""NOTE: Above analysis was performed from file "",14A";D_
file$
811    PRINT USING "10X,""       Output data are stored in file "",14A";Ofile$
820    END IF
830    ASSIGN @File TO *
840    END
850    DEF FNTvsv(Emf)
860    COM /Cc/ C(7)
870    Sum=C(0)
880    FOR I=1 TO 7
890    Sum=Sum+C(I)*Emf I
900    NEXT I
910    RETURN Sum
920    FNEND
```

APPENDIX H

SAMPLE CALCULATION

Data run number 5 (saturation temperature was 10 °C and heat flux was about 20 kW/m^2) was chosen for the sample calculation.

A. TEST-SECTION DIMENSIONS

D_o = 0.01588 (m)
D_i = 0.01270 (m)
D_1 = 0.01143 (m)
D_2 = 0.01588 (m)
L = 0.20320 (m)
L_u = 0.07620 (m)

B. MEASURED PARAMETERS

V_s = 3.58 (volts)
I_s = 2.27 (volts)
T_1 = 34.69 (°C)
T_2 = 30.25 (°C)
T_3 = 31.40 (°C)
T_4 = 33.49 (°C)
T_5 = 37.59 (°C)
T_6 = 31.38 (°C)
T_7 = 38.00 (°C)
T_8 = 32.96 (°C)
T_{sat} = 10.00 (°C)
k_c = 344.00 (W/m.K)

C. OUTER WALL TEMPERATURE OF THE BOILING TUBE

$p = \pi D_o = 3.1416 \times 0.01588 = 0.0499$ (m)

$A_c = \pi (D_o^2 - D_i^2)/4 = 3.1416 \times (0.01588^2 - 0.0127^2)/4$

$A_c = 714 \times 10^{-7}$ (m²)

$V = 25V_s = 25 \times 3.58 = 89.50$ (volts)

$I = I_s = 2.27$ (amp)

$Q = VI = 89.50 \times 2.27 = 203.17$ (W)

$$T_{avg} = \left(\sum_{n=1}^{8} T_n \right) / 8 = 33.72 \text{ (°C)}$$

$\bar{T}_{wo} = T_{avg} - Q_H(\ln(D_2 / D_1)/2 \pi L k_c)$

$\bar{T}_{wo} = 33.72 - 203.17(\ln(15.88/11.43)/2x3.1416x203.2x344)$

$\bar{T}_{wo} = 33.57$ (°C)

$T_b = \bar{T}_{wo}$

$\theta = T_b - T_{sat} = 33.57 - 10.00 = 23.57$ (°C)

D. PROPERTIES OF R-114 AT FILM TEMPERATURE {REF. 24}, {REF. 25}

$T_f = (\bar{T}_{wo} + T_{sat})/2 = 21.78$ (°C) $= 294.78$ (K)

$\log \mu (10^{-3} Ns/m^2) = -4.4636 + 1011.47/T_f$

$\mu = \exp(-4.4636 + (1011.47/294.78)) \times 10^{-3}$

$\mu = 356 \times 10^{-6}$ (Ns/m²)

$\rho = 36.32 + 61.14\ j^{1/3} + 16.42\ j + 17.48\ j^{1/2} + 1.12\ j^2$ (lb/ft³)

$j = 1 - T_f / T_c$

T_c = Critical Temperature = 753.95 (R)

$T_f = 294.78$ (K) $= 530.6$ (R)

$j = 1 - 530.60/753.95 = 0.296$

$\rho = 91.55\ (\mathrm{lb/ft^3}) = 1466.49\ (\mathrm{kg/m^3})$

$\nu = \mu / \rho = 2.428 \times 10^{-7}\ (\mathrm{m^2/s})$

$k\ (\mathrm{W/m.K}) = 0.0710 - 0.000261\ T_f \quad (T_f \text{ in } {}^\circ\mathrm{C})$
$k = 0.0653\ (\mathrm{W/m.K})$

$C_p\ (\mathrm{kJ/kg.K}) = 0.4 + 1.65 \times 10^{-3} T_f + 1.51 \times 10^{-6} T_f^2 - 6.68 \times 10^{-10} T_f^3 \quad (T_f \text{ in K})$
$C_p = 1002\ (\mathrm{J/kg.K})$

$\alpha = k/\rho C_p = 4.444 \times 10^{-8}\ (\mathrm{m^2/s})$

$\beta = -(\Delta\rho / \Delta T)/\rho$
$\beta = -(1465.95 - 1466.49) / ((294.98 - 294.78) \times 1466)$
$\beta = 0.00184\ (1/\mathrm{K})$

$Pr = \nu / \alpha = 5.464$

E. HEAT-FLUX CALCULATION

Average natural-convection heat-transfer coefficient at non-boiling ends:

$$\bar{h} = \frac{k}{D_o}\left\{0.60 + 0.387\,\frac{\left[\dfrac{g\beta D_o^3 \theta_B \,\mathrm{Tanh}\left\{\left(\dfrac{\bar{h}P}{k_C A_C}\right)^{1/2} L_u\right\}}{\nu\alpha L_u \left(\dfrac{\bar{h}P}{k_C A_C}\right)^{1/2}}\right]^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}}\right\}^2$$

$\bar{h} = 282\ (\mathrm{W/m^2.K})$

Heat-transfer rate through non-boiling ends:

$Q = (\bar{h}\ p\ k_c A_c)^{1/2}\ \theta_b \mathrm{Tanh}(mL_u)$

$m = (\bar{h}\ p\ /\ k_c A_c)^{1/2} = 23.94\ (1/m)$

$Q_F = 13.15\ (W)$

$Q_{Loss} = 2 \times Q_F = 2 \times 13.15 = 26.30\ (W)$

Heat flux through active boiling surface:

$Q = Q_H - Q_{Loss} = 203.17 - 26.30 = 176.87\ (W)$

$q = Q\ /\ A_b$

$A_b = D_o L = 10.137 \times 10^{-3}\ (m^2)$

$q = 176.87\ /\ 0.010137 = 17447.96\ (W/m^2)$

$h = q\ /\ \theta_b = 740.26\ (W/m^2.K)$

The following are the results obtained from the computer by running the data reduction program (See Appendix F).

$q = 17490\ (W/m^2)$

$\theta_b = 23.57\ (°C)$

$h = 742\ (W/m^2.K)$

APPENDIX I

UNCERTAINTY ANALYSIS

The same data set (run number 5) that was used for the sample calculation was chosen for the uncertainty analysis; therefore, the dimensions of the test section, and the measured and calculated parameters found in the sample calculation were used in this analysis. All uncertainties are presented as a percentage of the calculated parameter.

A. UNCERTAINTY IN SOURCE HEAT-TRANSFER RATE

Q_H = VI (W)

$I = I_s = 2.27$ (amp) $\qquad \delta I = \pm 0.025$ (amp)

$V = 25V_s$; $V_s = 3.58$ (volts) $\qquad \delta V_s = \pm 0.05$ (volts)

where:

δ = uncertainty in measurement and calculation

$$\delta Q_H/Q_H = ((\delta V_s/V_s)^2 + (\delta I/I)^2)^{1/2}$$

$$\delta Q_H/Q_H = ((0.05/3.58)^2 + (0.025/2.27)^2)^{1/2}$$
$$= 1.80 \text{ percent}$$

B. UNCERTAINTY IN SURFACE AREA

$A_b = \pi D_o L$

$D_o = 15.88$ (mm) $\qquad \delta D_o = 0.1$ (mm)

$L = 203.20$ (mm) $\qquad \delta L = 0.1$ (mm)

$$\delta A_b/A_b = ((\delta D_o/D_o)^2 + (\delta L/L)^2)^{1/2}$$

$$\delta A_b/A_b = ((0.1/15.88)^2 + (0.1/203.2)^2)^{1/2}$$
$$= 0.63 \text{ percent}$$

C. UNCERTAINTY IN WALL SUPERHEAT

$$\Delta T = T_{wo} - T_{sat}$$

$$T_{sat} = 10.00\ (°C) \qquad \delta T = 0.5\ (°C)$$

$$T_{wo} = T_{avg} - Q_H (\ln(D_2/D_1) / 2\pi L k_c)$$

$$T_{avg} = \left(\sum_{n=1}^{8} T_n\right) / 8$$

where:

T_n = thermocouple readings (See Appendix G)

$T_{avg} = 33.72\ (°C)$

$$S.D. = ((\sum_{n=1}^{8}(T_n - T_{avg})^2)/7)^{1/2} = 2.687\ (°C)$$

where:

S.D. = standard deviation

Since logarithmic term in equation of "$\bar{T}_{wo}$" is too small when compared to standard deviation, this term can be neglected for uncertainty analysis (i.e., $\bar{T}_{wo} = T_{sat}$).

$$\bar{T}_{wo} = 33.72\ (°C) \qquad \delta\bar{T}_{w0} = 2 \times S.D. = 5.37\ (°C)$$

$$\Delta T = \bar{T}_{wo} - T_{sat} = 33.72 - 10.00 = 23.72\ (°C)$$

$$\delta\Delta T/\Delta T = ((\delta\bar{T}_{wo}/\Delta T)^2 + (-\delta T_{sat}/\Delta T)^2)^{1/2}$$

$$\delta\Delta T/\Delta T = ((5.37/23.72)^2 + (-0.5/23.72)^2)^{1/2}$$

$$\delta\Delta T/\Delta T = 22.7 \text{ percent}$$

D. UNCERTAINTY IN HEAT FLUX

$$q = (Q_H - 2Q_F)/A_b$$

$$Q_H = 203.17\ (W) \qquad \delta Q_H = 3.65\ (W)$$

Assuming the same proportion in the uncertanity for Q:

$$Q_F = 13.15\ (W) \qquad \delta Q_F = 0.23\ (W)$$

$$Q_H - 2Q_F = 176.87\ (W)$$

$$\delta q/q = \left((\delta Q_H/(Q_H - 2Q_F))^2 + (2\delta Q_F/(Q_H - 2Q_F))^2 + (\delta A_b/A_b)^2 \right)^{1/2}$$

$$\delta q/q = \left((3.65/176.87)^2 + (0.46/176.87)^2 + (-0.0063)^2 \right)^{1/2}$$

$$= 2.17 \text{ percent}$$

E. UNCERTANITY IN BOILING HEAT-TRANSFER COEFFICIENT

$$h = q/\Delta T$$

$$\delta h/h = \left((\delta q/q)^2 + (-\delta\Delta T/\Delta T)^2 \right)^{1/2}$$

$$\delta h/h = \left((0.0217)^2 + (-0.227)^2 \right)^{1/2}$$

$$= 22.8 \text{ percent}$$

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